PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

The Big Job That's Coming

RECENTLY IN AN INFORMAL DISCUSSION Dr. C. E. Lucke of Columbia University made a statement to the effect that "It is of greatest importance to convince the engineering student that the same thing can be accomplished in a great number of ways. Engineering is not rigid; there are always many different materials, many different mechanisms and many different types of constructions that might be used. Each would result in a satisfactorily operating machine that will do the work required. But commercial considerations make it of greatest importance that the results be achieved at lowest reasonable cost, otherwise the competitors get the business.

"Of course," continued Dr. Lucke, "just now war conditions are dominating, at least to a large extent, and for some products costs do not have the importance they have in peace time. But we will not always be at war; peace will come and then competition will be restored to the peace time basis."

No one can find fault with Dr. Lucke's views; in them there is much food for thought. When peace comes the engineering design picture will be entirely different from anything it has ever been. During this period of stress, many new developments will come to pass. By force of circumstances, design and production engineers will have learned in greatest detail about the properties and processing of many materials they had never even considered before. They will have become familiar with the hidden virtues of both strange and common materials and how to circumvent their deficiencies and peculiarities.

Another revolution in the making is the price structure in the raw materials markets. Huge quantity production of hitherto "high cost" materials will result in lowering their price materially. Aluminum, synthetic rubber-like materials, plastics, and also perhaps new materials as yet unnamed, will be available at prices that will widen their fields of application immeasurably.

Add to all these economic changes that are bound to occur in the prices of raw materials, the new developments that will inevitably be brought forth from scientific research laboratories during the next few years, the possibilities appear staggering. Whereas the design engineer today may have ten ways of achieving the desired result, he will have hundreds when the present stress is over and necessity, the mother of invention, has brought forth prolifically another brood of new materials and new ways of doing things.

To keep abreast of this great wave of technological progress, minds must be alert and the quest for knowledge continuous. And those things which today hold the greatest attention of the designers will be the things incorporated in the products of the days which we now call the post-war period.

Already design engineers are dreaming, scheming and even laying out designs of products to be made when peace comes. A few companies have gone so far as to develop complete designs, details and specifications included, of products to be made when the wars are over. Others have made tentative plans. But all are awake to the fact that there is a big job in the making—a task that will tax the ingenuity of all design engineers.

Packard Clipper being driven at high speed through propeller backwash from Army pursuit planes. By special instruments attached to steering mechanism, Packard engineers were able to determine the car's relative stability in a cross wind

AUTOMOTIVE DESIGN IN THE 1942 CARS

DESPITE A PRIORITY situation which severely limited new design and which necessitated considerable alterations in formerly standardized parts, automotive engineers have succeeded in bringing out new models for 1942. For obvious reasons mechanical changes have been kept to a minimum, but exterior appearance has been greatly improved—the cars are more streamlined than ever. This is possibly due to the anticipation that dies may not be available for '43 changes, hence styling advances this year are more radical than it has ever been before.

Contrary to general opinion, the new cars have extensive chromium plate, and stainless trim, explained by the fact that many of the companies have considerable fabricated stock on hand and intend to use it as long as it holds out. It is expected that in the later months of the model-year chromium trim will gradually disappear and will be replaced by painted strip or extruded plastics. Oldsmobile is tentatively experimenting with aluminum paint for such parts. Zinc die-cast grilles have been entirely replaced by steel stampings. This change necessitated a tremendous increase in cost but had to be made, as some cars carried as much as 30 lb. of zinc in the radiator grille alone. One manufacturer is experimenting with chromium plated plastic, injection molded, for radiator grilles; this is strictly in the laboratory stage; many difficulties, such as bond between plating and plastic, fastening means, expansion and contraction difficulties, and molding problems, must be overcome before this design is practical. It was found that the chromium plate added a slight amount of stiffness to the finished molding.

Cast iron piston developed for series 40 Buick engines. Special foundry technique was required to cast the piston with the "fire-ball" top, which needs no machining

The three materials automotive engineers had to "design-out" for '42 were: aluminum, zinc, and nickel steel. A great many companies had been using

aluminum pistons and the change to a cast-type piston brought problems of lower thermal conductivity, reduction to half the thermal expansion and increased reciprocating weight. Fortunately, most of the manufacturers last year had adopted thin-babbit bearings, backed by steel or by powdered iron matrix. These bearings have sufficient fatigue strength to take the increased load caused by cast pistons. Most of the plants have prepared plans to produce cast-iron pistons; through improvements in foundry practice sections have been reduced considerably. Buick with its "fire-ball" top piston had a particularly difficult foundry problem to solve, but this piston is now cast iron on the series 40 and no machining of the top is required. Oldsmobile has developed a piston design in which the piston is cast of ArMaSteel, a pearlitic malleable iron with an arrested heat-treatment containing sufficient carbon to be classified as a steel. This piston weighs 48 percent more than former aluminum piston as compared with the 80 percent increase in weight of a cast-iron piston over aluminum. Consequently there is considerable reduction in stress on the connecting rod and bearings. To be on the safe side, however, the connecting rod in the Oldsmobile engines has been made about 8 percent heavier and the crankshaft has been somewhat increased in weight.

Buick was experimenting with so-called "roughened" bearing surfaces and have found that for their engine this "roughened" surface gives a considerable increase in fatigue life. The surface is ground and then lightly lapped to remove ridges and it is reasoned that the remaining pockets act as reservoirs for minute amounts of oil. Pontiac is continuing the use of cast-iron pistons, tin plated. The tin normally aids in the breaking in of the piston. Others makers such as Packard and Studebaker are Parco-Lubrizing their cast-iron pistons. This process produces an oxide coating which tends to hold oil and insures smooth operation.

Die-Cast Parts

Zinc die-castings have been cut down considerably in number. Many parts such as door handles, ornaments, horns, are being made of stamped steel or of plastic, injection molded over a steel core. Radiator ornaments on Packard cars are die-cast of antimony lead, a good substitute for zinc in cases where structural strength is not required. Other ornaments on cars such as Pontiac are of acrylic plastic. Packard's transmission cover, formerly of zinc, is now cast-iron. Typical alterations in materials and methods of manufacture are given in the table below.

Automatic Transmissions

Automatic transmission are becoming increasingly popular with designers. As an example of their success, Oldsmobile sold more than 48 percent of their 1941 cars with automatic transmissions, even at the $90 premium. This year Oldsmobile and Cadillac will continue with the Hydramatic transmission described in PRODUCT ENGINEERING, November 1939, and Chrysler will continue the simplimatic transmission described in October 1940, PRODUCT ENGINEERING. The Oldsmobile drive is essentially a set of planetary gears driven through a hydraulic coupling, with shifting controlled by a centrifugal governor. The transmission is completely automatic. The Chrysler drive incorporates a fluid coupling and vacuum-controlled, semi-automatic transmission to which overdrive can be added. This year three other makers have adopted semi-automatic transmissions to their lines; Ford, Studebaker, and Hudson. One other maker tentatively considered an automatic transmission but the supplier was unable to guarantee manufacture.

Hudson's automatic drive, described on pages 518-520 of this article, consists in an arrangement of vacuum cylinders and linkages, electrically governed. This is a further refinement of their automatic clutch of previous years. The Studebaker transmission is also vacuum actuated and electrically controlled, but includes a fluid coupling and clutch combination. This drive is described on pages 522-524.

Although basic mechanical changes in the cars have been relatively few, many minor refinements have been made in design to increase efficiency. These changes, although minor to the average automobile, often show significant trends in engineering.

Considerable credit must go to automotive engineers because of the feat they have accomplished in readying for production new cars while at the same time taking on more and more responsibility for the production of armored equipment. It was no small job to make new refinements, change methods and materials on formerly standard parts, set up the 1942 production program and at the same time put tremendous effort towards the tooling up and production of tanks, airplane parts, guns, armored trucks and airplane engines. The automotive industry has been able to do this only because of its skill of the designers and their tremendous production experience.

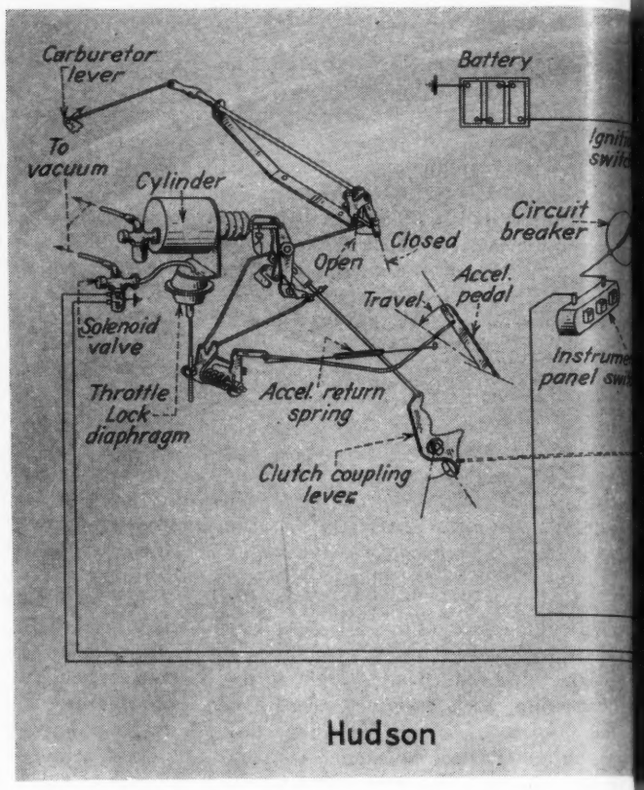
TYPICAL ALTERNATES IN 1942 CARS

PART	FORMER MATERIAL	ALTERNATE MATERIAL
Axle Shafts	X-3150*A*	4150
Rear Axle Ring Gear	4615	5120
Steering Knuckles	3135*A*	4140
King Pins	3115*A*	4115
Rear Axle Gearing	4600	1320
Trim	18-8 Stainless	18 Cr Stainless
Exhaust Valves	Cr Ni (1½-12%Ni)	7-18%Cr 2-3½%Si
Inlet Valves	3140	4140
Castings	Fe + 0.25—1.25% Ni	Fe + Si or + Cu
Connecting Rod	1045	X1345
Pistons	Secondary Aluminum	Cast Iron Cast Steel
Transmission Gears	Nickel Steel	Amola Steel
Timing Gear	Aluminum	Steel & Plastic
Cylinder Heads	Aluminum	Cast Iron
Brake Wheel Piston	Aluminum	Powdered Metal
Valve Chamber Cover	Aluminum	Cast Iron
Door Handles	Die-Cast Zinc	Stamped Steel Plastic and Steel
Grilles	Die-Cast Zinc	Stamped Steel
Ornaments	Die-Cast Zinc	Die-Cast Antimony Lead Plastic
Steering Post Hub	Die-Cast Zinc	Steel Stampings Malleable Casting
Instrument Panel Grille	Die-Cast Zinc	Plastic
Radiator Shell	Brass	Copper
Reflector (Lamp)	Silver-Plated Brass	Glass
Speedometer Gear	Die-Cast Zinc	Plastic
Horn Ring	Die-Cast Zinc	Stamped Steel
Transmission Cover	Die-Cast Zinc	Cast Iron Plastic & Steel
Shift Control Arm	Die-Cast Zinc	Forging
Front Suspension Arms	Forgings	Stampings
Bushings	Solid Bronze	Oilite on Steel
Gaskets	Copper-Asbestos Cork	Steel-Asbestos Ligno-Neoprene
Horns	Die-Cast Zinc	Steel Stampings

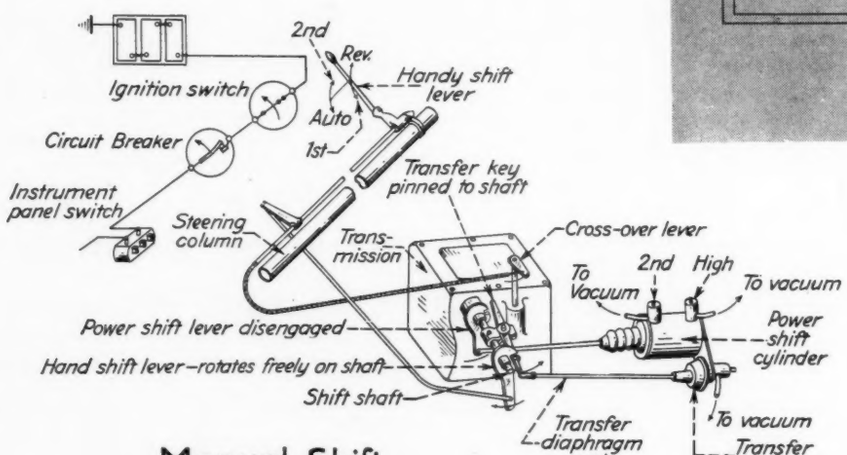
Hudson's Electrically-Controlled Vacuum-

THROUGH THE USE of an arrangement of electrically-controlled vacuum power units operating through mechanical linkages, the Hudson Drive-Master transmission eliminates clutch operation and gear shifting except when reversing or starting under unusual conditions. With the gear shift lever in high gear position, depressing the accelerator pedal moves the car forward in pickup gear, equivalent to the usual second speed gear. To go into high gear it is only necessary to release the accelerator. When brakes are applied the transmission is automatically shifted from high to pickup gear and the car is ready for a new start. There is no tendency for the car to "creep" or move forward in neutral with the engine running and, therefore, no necessity to continually apply brakes when idling at traffic intersections.

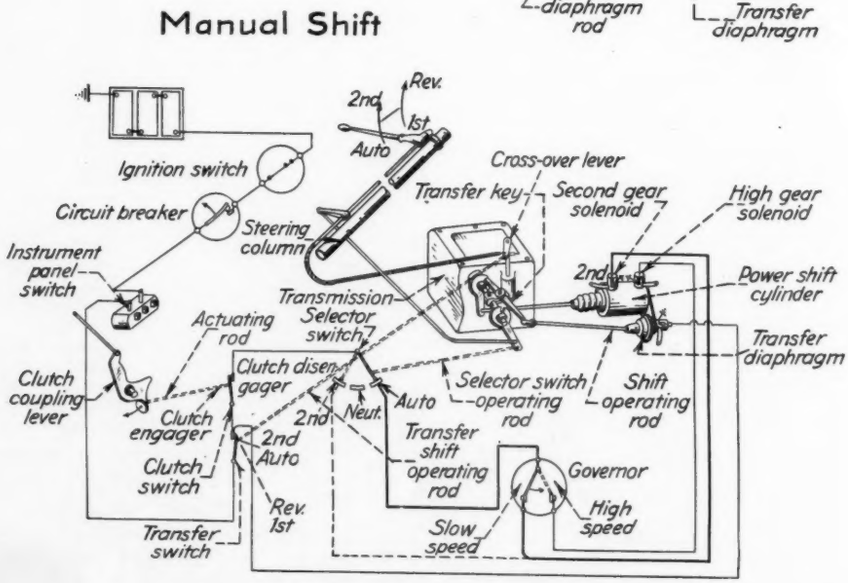
Three driving methods are available: normal driving without gear shifting or clutch operation, manual gear shifting without using the clutch, or conventional gear shift and clutch operation. These options are selected by pushbuttons on dashboard. Regardless of drive method being used, gears can be shifted manually any time the driver so desires.



Hudson



Manual Shift



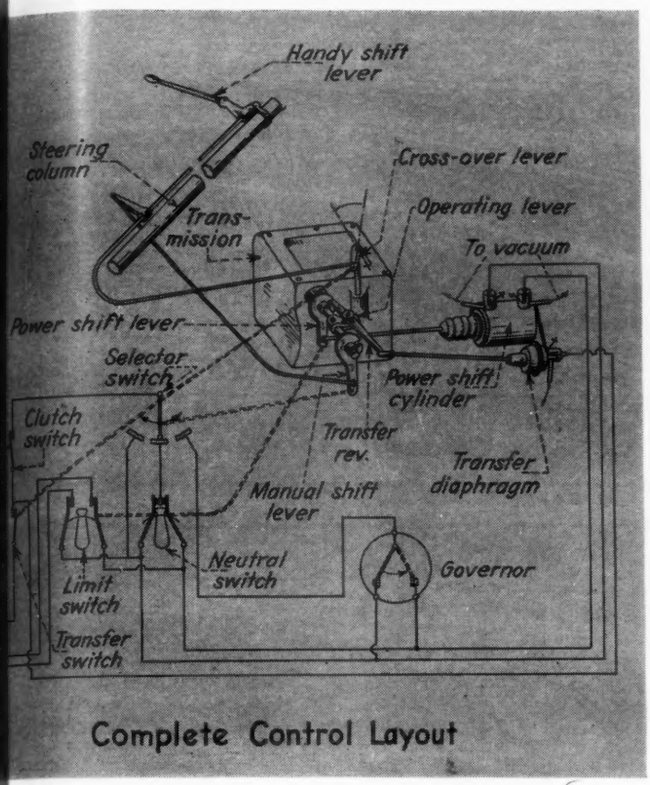
Power Shift Control

Conventional operation. The desired gear is selected by lever on steering arm which actuates the hand shift on the transmission. For manual operation this lever is mechanically locked to the shift shaft by a transfer key pinned to transfer key hub that is in turn secured to the shaft by setscrew. Transfer diaphragm rod holds transfer key engaged in hand-shift lever by spring pressure.

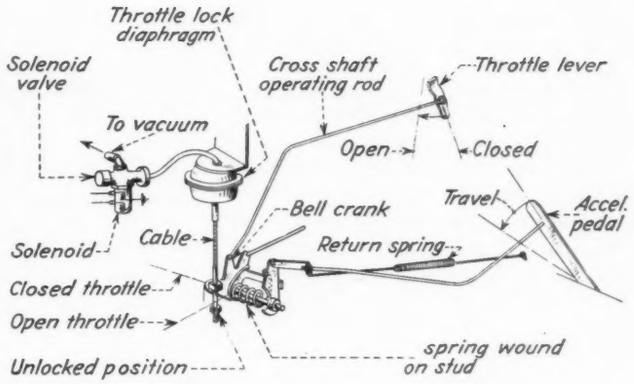
Power shifting. Current from the battery to the transfer switch is supplied by pressing in the "HDM" button on the instrument panel. The circuit from the transfer switch to solenoid valve is completed when the transfer switch rod is operated by cross over lever as the steering shift arm is moved through neutral to the second-high side. With the transfer switch closed, the solenoid valve is energized and vacuum is admitted to transfer diaphragm. This pulls rod backward, disengaging transfer key from hand shift lever and engaging it with power shift lever.

The system is now ready for power shifting. A governor switch, driven mechanically by the transmission gears

Operated Clutch and Transmission



Complete Control Layout

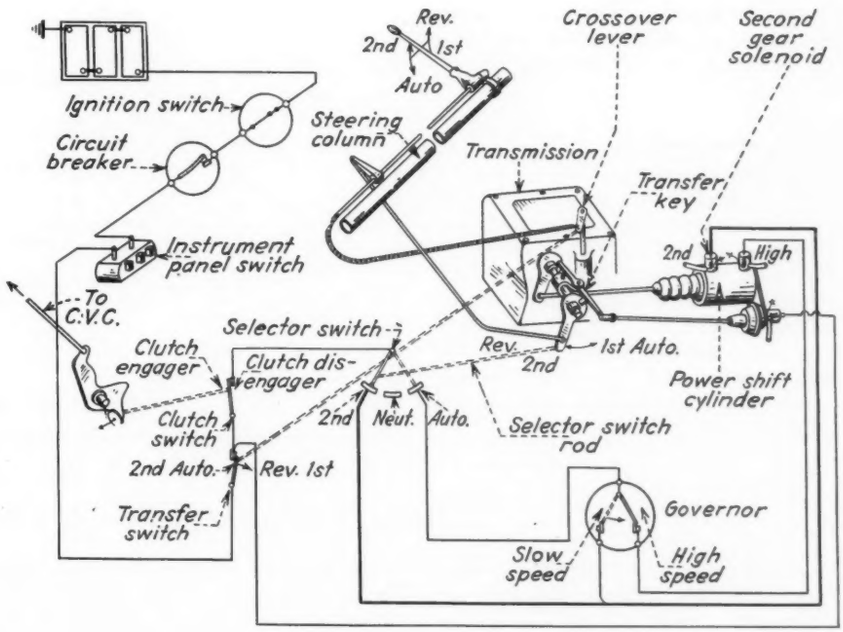


Throttle Locking Device

Throttle-locking device, shown in unlocked position. To prevent "beating the shift"—applying power before the shifting of gears has been completed. Throttle lever on carburetor is actuated by a bellcrank which is in turn connected to the accelerator pedal linkage through a torsion spring. When shifting is completed, throttle lock diaphragm releases throttle lever bellcrank and accelerator pedal pressure actuates throttle bellcrank by means of the spring. If shift is not completed, the diaphragm holds the bellcrank in a closed position; accelerator pedal pressure cannot actuate the bellcrank, but merely winds up the torsion spring.

and fitted with two sets of contacts—one set for slow-speed operation and the other for high-speed operation, controls operation of the clutch-shifting which is actuated through engine vacuum. When the clutch is disengaged, the clutch switch is closed through movement of clutch coupling lever and operating rod, and the circuit completed through the automatic side of selector switch to the slow speed contacts of governor switch and to second gear solenoid.

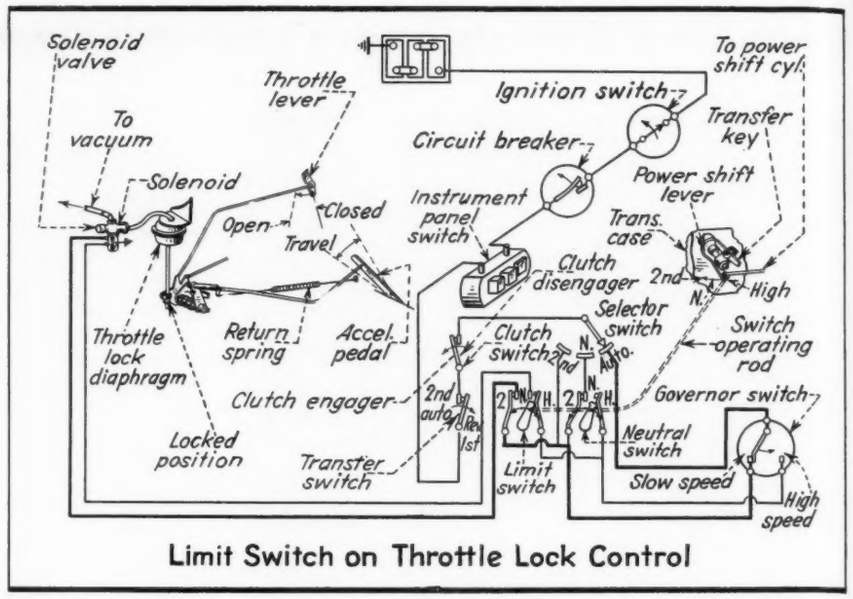
Power shift into second gear. As the second gear solenoid is energized, vacuum is admitted to the front end of the transmission shift cylinder, pushing operating rod forward, and rotating power shift lever into second gear position. Since the transfer key is engaged, transfer key hub and shaft rotate causing the gears to shift into second speed. With shift completed and clutch engaged, the circuit from clutch switch to second gear solenoid is broken, deenergizing the solenoid. If the car is operating at slow speeds, the circuit to the second gear solenoid is completed; if the car is operating at high speed, the contacts in governor switch move to high-speed position and close the circuit to high gear solenoid, causing shift into high gear.



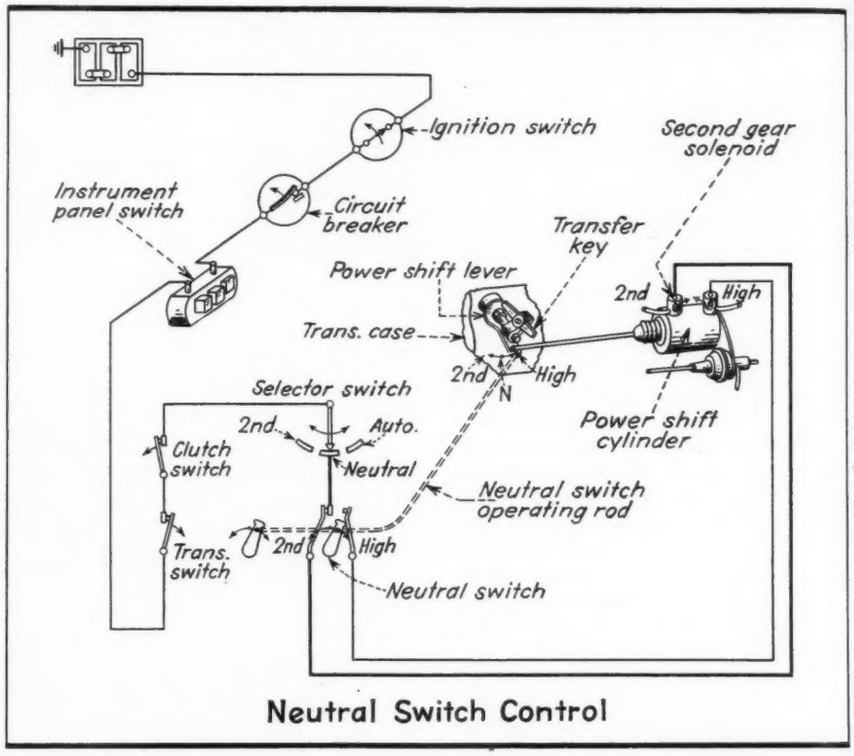
Governor By-Pass for Shift Down

Shift down to second gear. For drop into second gear while car operates above high gear shifting speed, a positive circuit is used to by-pass the governor switch. The lever on steering post is thrown into second gear position, moving selector switch rod and closing second gear contacts in selector switch thus completing the circuit to second gear solenoid. The by-pass circuit is shown in heavy lines in the sketch. (*Continued on next page*)

Whenever a power shift is made, the solenoid, energized, admits vacuum to throttle lock diaphragm, pulling up on diaphragm cable until the two lock nuts attached to the lower end of the cable contact the swivel block. If the accelerator pedal is depressed when the throttle lock is in this position, the pressure exerted merely tends to wind up the bellcrank spring. The diaphragm is always energized while the shift is being made and remains so until the shift is completed. A limit switch which contains two sets of contacts governs the action of the throttle lock solenoid. The switch is actuated by the neutral switch operating rod attached to the power shift lever on the transmission. For example, with the transmission in the high gear position and the car operating below the high gear shifting speed, the governor switch will call for a shift to second gear. The high gear contacts in the limit switch will be separated and the second gear contacts closed. The circuit will be completed to one of the windings in solenoid valve, energizing the solenoid, admitting vacuum and pulling the throttle lock cable upward into the locked position. With the shift to second gear completed, the cam in the limit switch opens the second speed contacts de-energizing solenoid valve, permitting the throttle to open. As the limit switch cam passes through the neutral position on its way to second gear position, the high gear contacts close, preparing the circuit for a shift into high gear.



Limit Switch on Throttle Lock Control



Neutral Switch Control

Neutral switch. To stop the shift in neutral when going into low or reverse gear, another switch is actuated by a switch rod connected to the transmission power shift lever. When the selector switch is moved into neutral, the circuit is completed to the proper gear solenoid and a power shift toward second gear is started. As the shift reaches neutral position, a cam on the neutral switch separates the contacts, thus stopping the shift in neutral.

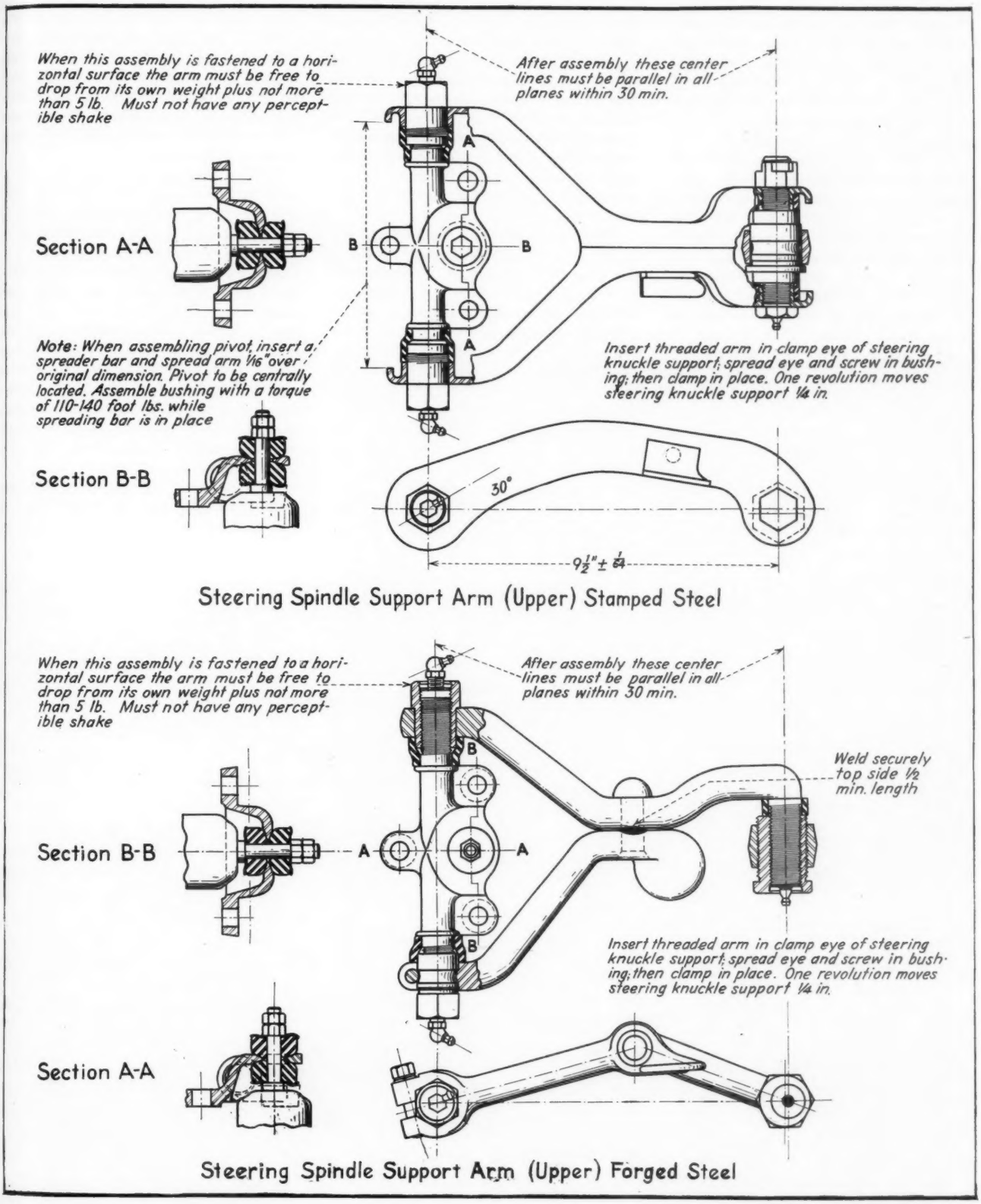
SOME ADDITIONAL HUDSON CHANGES

TO SECURE QUIETER TIMING GEAR OPERATION, back lash between the teeth of Hudson crankshaft and camshaft gears has been reduced to between 0.002 in. and 0.004 in. This is about half the clearance formerly used; the change has made a material improvement in quietness of the front end drive.

RUBBER BUMPERS which were mounted on the rear axle housing directly under the frame side members have been discontinued and in their stead new rectangular shaped rubber bumpers are now mounted on the lower flange of the frame side members in the 1942 Hudsons. These bumpers prevent metal-to-metal contact in extreme flexing of the springs under unusual conditions. A new rear axle differential carrier bumper is attached to the lower flange of the frame cross member immediately above the axle housing. This bumper is also of molded rubber.

TO MINIMIZE OIL PUMP "STARVATION," a change has been made in the design of the oil reservoir suction pipe used on both six and eight-cylinder Hudson engines. Previously the lower end of the pipe became uncovered when the oil banked against the side of the reservoir when the car was making turns at high speed. With the new pipe in the center of the oil reservoir, the lower end remains submerged in oil even when oil level is low.

Alternate Designs of Control Arms


When this assembly is fastened to a horizontal surface the arm must be free to drop from its own weight plus not more than 5 lb. Must not have any perceptible shake
After assembly these center lines must be parallel in all planes within 30 min.
Section A-A
Note: When assembling pivot, insert a spreader bar and spread arm 1/16" over original dimension. Pivot to be centrally located. Assemble bushing with a torque of 110-140 foot lbs. while spreading bar is in place
Section B-B
Insert threaded arm in clamp eye of steering knuckle support; spread eye and screw in bushing; then clamp in place. One revolution moves steering knuckle support 1/4 in.
30°
9 1/2" ± 1/64
Steering Spindle Support Arm (Upper) Stamped Steel
When this assembly is fastened to a horizontal surface the arm must be free to drop from its own weight plus not more than 5 lb. Must not have any perceptible shake
After assembly these center lines must be parallel in all planes within 30 min.
Weld securely top side 1/2 min. length
Section B-B
Insert threaded arm in clamp eye of steering knuckle support; spread eye and screw in bushing; then clamp in place. One revolution moves steering knuckle support 1/4 in.
Section A-A
Steering Spindle Support Arm (Upper) Forged Steel

Two types of front suspension upper control arms are being used in regular production on 1942 Hudsons. One is drop-forged type arm as used on the 1941 models and the other is constructed of heavy steel stampings similar to those used in the lower control arm. In the pressed steel type, the upper end of the spindle support is carried on an eccentric bushing mounted on a pivot bolt which divides the load uniformly to both sides of the arm.

Studebaker's Semi-Automatic Transmission

A FLUID COUPLING with an automatic clutch and overdrive transmission are the essentials of the new Studebaker "Turbo-Matic" drive, available as extra-cost equipment. Feature of the new drive is the elimination of the clutch pedal. The driver, through the shift lever on the steering column, can choose the gear ratios best suited to traffic conditions. To eliminate the possibility of starting the engine in gear, a switch on the shift control prevents operation of the starting motor unless the shift lever is in neutral position. Reverse gear is used as in the past. Low is used only for tough pulling and in starting on steep grades. Second gear is used for most starts, for stop-and-go driving and general traffic conditions. Third is used for light traffic or cruising, and overdrive is used in both "traffic" and "cruising" ranges. Car starts in gear ratio selected and when speed reaches more than 15 mi. per hr., the transmission goes into overdrive upon release of the accelerator for an instant. To return to conventional the accelerator is depressed to the floor. All starts are made in conventional gear.

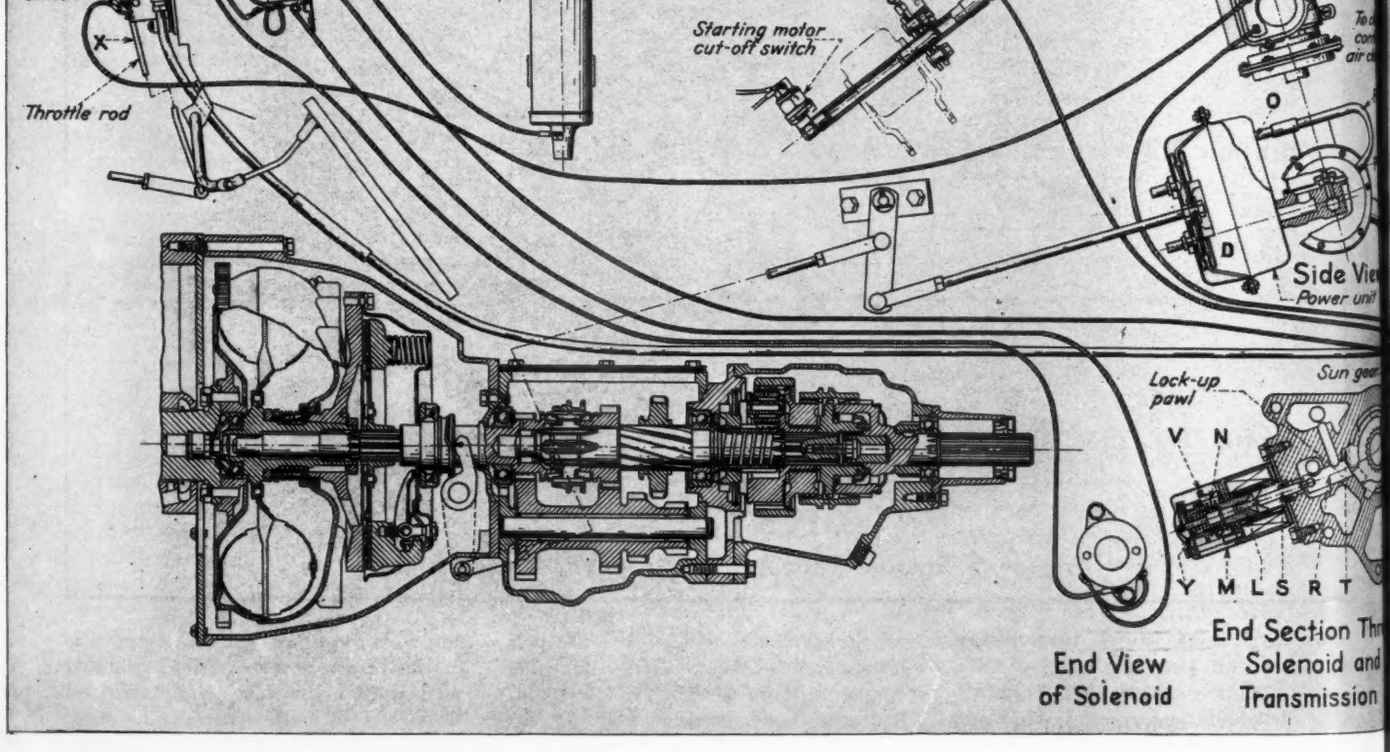
Clutch release and engagement is effected by a frame-mounted power unit consisting of a diaphragm upon which engine manifold vacuum can act to cause clutch release, a solenoid which controls the admission of engine manifold vacuum, and a balancing valve to provide a cushion for clutch engagement.

When the power unit solenoid is energized (shown at de-energized position on diagram), plunger *A* is raised, thus operating a valve which closes passages *B* to the balancing valve and *K* to the throttle control and also opens the passage *C* to the engine manifold. Engine manifold vacuum is thus admitted to declutching side *D* of the power unit diaphragm and clutch is released. The diagram shows power unit diaphragm at position of clutch engagement. A simple system of rods and levers connects the power unit to the clutch release shaft.

Two separate electrical circuits control the power unit solenoid, hence clutch release. Neither circuit to the solenoid can be completed unless the throttle is closed. Closing the throttle does not release the clutch, but the throttle must be closed before clutch release can occur. These circuits are *I* through the governor, and *II* through the over-rule switch.

I—Governor driven from the transmission speedometer drive gear, has two sets of contact points (governor is shown at zero speed on diagram). One set is for one of the power unit solenoid circuits. This set of points is in contact until a car speed of about 15 mi. per hr. is reached, at which speed the contact is broken thus interrupting the circuit to the power unit solenoid. On speed decrease, the governor contacts "make" at about 6 mi. per hr. With the throttle closed at car speeds below 6 mi. per hr. the circuit to the power unit solenoid will be completed and the clutch released.

II—The other circuit is through the over-rule or selector switch mounted on the steering column and actuated by



Combined With Fluid Coupling

the gear shift controls. Initial movement of the gear shift lever completes the circuit, releasing the clutch before the actual shifting of gears is started. The circuit is broken by a return spring as soon as movement of the shift lever ceases.

Clutch Engagement

When the power unit solenoid shown below is de-energized, plunger *A* is returned downward by spring pressure. This closes off engine vacuum to the clutch control diaphragm and opens passage *B* to the *D* side of the clutch release diaphragm. Passage *F* admits the vacuum present in the power unit to the *G* side of the cushion valve diaphragm. Atmospheric pressure acts against the *H* side. As long as the difference between atmospheric pressure *H* and vacuum *G* exceeds the tension of spring *J*, valve *E* will remain open and admit air to the power unit. This action cannot take place unless plunger *A* is at the down or de-energized position. With plunger *A* pulled upward, the *B* passage is closed and air cannot be admitted to the system.

The cushion valve serves to balance the vacuum in the power unit to the setting of the cushion valve spring. This setting is adjustable through a screw and is set at about 4½ in. mercury. With this amount of vacuum in the power unit, the clutch is just engaged. The cushion valve action is not instantaneous because of restrictions at points *O*, *P* and *Q*, hence initial clutch engagement is gradual.

The above occurs if the throttle remains closed. Under normal driving conditions, the throttle will be opened gradually or quickly at the option of the driver. This will supplement clutch engagement through a control in the throttle mechanism.

Conditions under which the power unit solenoid is energized:

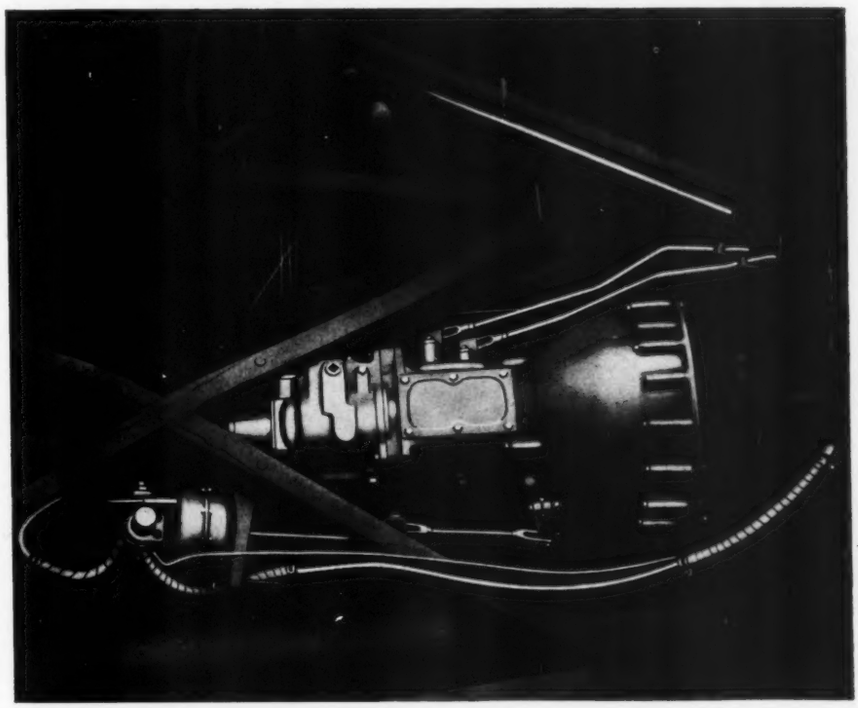
1. Transmission governor closes circuit at car speeds below 6 mi. per hr.
2. Over-rule switch on gear shift closes circuit whenever movement of gear shift lever is started.

Conditions under which the solenoid unit is opened, solenoid de-energized:

1. Initial movement of throttle (accelerator pedal) from closed position breaks the circuit regardless of other controls.
2. Above 15 mi. per hr., transmission governor opens the circuit.
3. Whenever movement of gear shift lever is stopped, the circuit through the over-rule switch is opened.

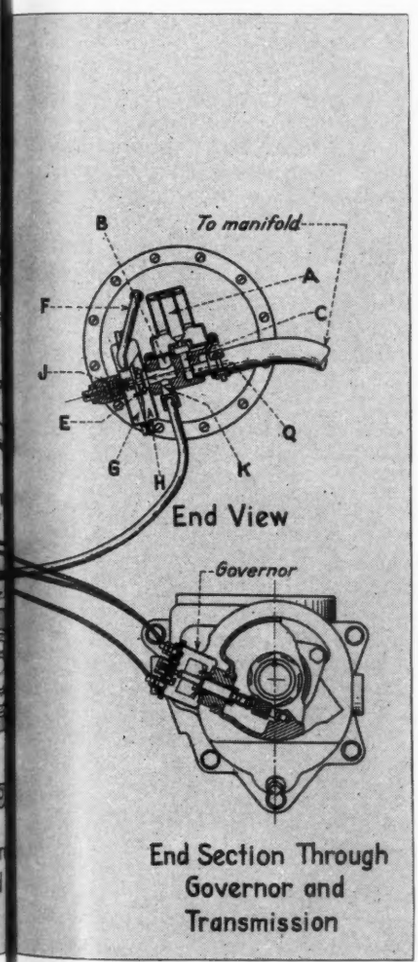
At speeds below 6 mi. per hr., clutch engagement can occur only as the throttle is opened.

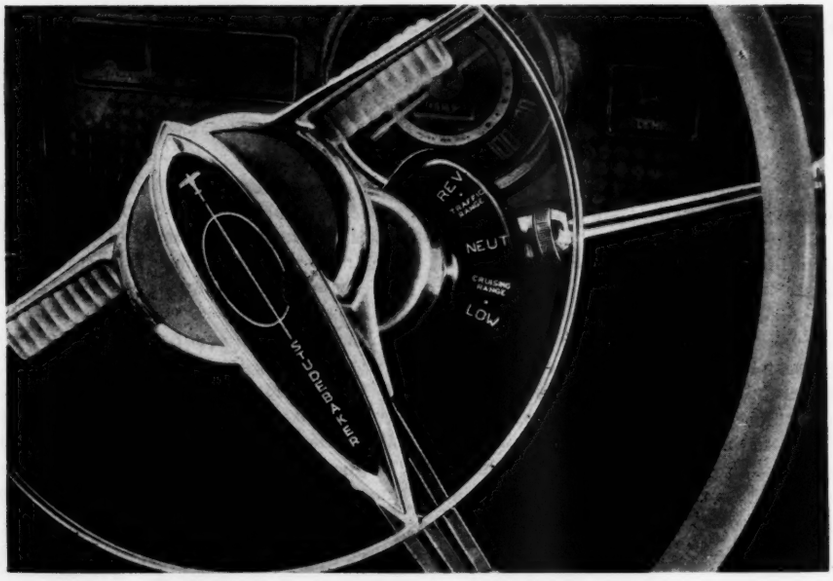
A dash-mounted combination air control valve and switch is connected in the throttle linkage. Initial movement of the control lever (3-4 deg.) breaks a pair of contact points, interrupting the electrical circuit to the power unit solenoid, thus starting action of the power unit cushion valve. At between 7 and 8 deg. movement of the lever, the air valve starts to open admitting atmospheric pressure to the passage *K* of the power unit and thence to the *D* side of the power unit diaphragm. This modifies the action of the cushion valve. Slow or quick clutch engagement is available to the driver, dependent on how fast or far he pushes the accelerator pedal. On slow starts, the accelerator pedal is pushed down gradually and clutch engagement is also gradual. On



(Above) Compact design of Studebaker fluid transmission is obvious from this view

(Left) Mechanical and electrical design details of Studebaker vacuum-controlled, semi-automatic transmission. Parts are shown in proper relative positions





Control of the Studebaker semi-automatic transmission is through selector shift mounted on steering column

fast starts, the accelerator pedal is pushed down quickly and fast clutch engagement follows.

Continuing movement of accelerator pedal would result in over-travel of the throttle control unit lever, except for the presence of a friction slip joint between the operating rod and the lever. As the throttle is returned to the closed position, this slip joint automatically returns the air valve of the unit to the closed position and closes the electrical contact points.

Normal clutch engagement is governed by power unit cushion valve action modified by the rate and extent of throttle opening. However, at speeds above 15 mi. per hr. clutch engagement will occur as the gear shift is completed even though the throttle is not again opened. Clutch engagement under this condition will be slower than usually dictated by the driver through throttle operations.

On initial speed increase, the governor points "break" at approximately 15 mi. per hr. but do not again "make" with speed decrease until about 6 mi. per hr. Hence, after having reached a speed of above 15 mi. per hr. the governor will start action of the power unit cushion valve after completion of a shift down to about 6 mi. per hr.

Starting Motor Cut-Out Switch

To prevent starting the engine with the transmission in gear, there is mounted on the lower end of the gear shift controls a switch in the starting motor relay circuit which prevents operation of the starting motor unless the gear shift lever is at "neutral" position. A key on gear shift shaft depresses a ball in the switch only when shift lever is at "neutral." This action closes switch contacts and permits operation of the starter motor.

Transmission

The transmission, in conjunction with the automatic clutch, is semi-automatic in action and requires a minimum of gear shifting. Two ratios under direct driver-control are available in each range. The forward gear case is similar to that used with the overdrive transmission on prior models. Helical gears are used throughout with three forward ratios. Synchronizer and blocker ring action assure quiet gear action in both second and direct ratios. The overdrive gearing contained in the rearward case is similar to that previously used. In place of overdrive engagement by centrifugally actuated pawls, the gearing is in the overdrive ratio at all times. A single pawl actuated by a solenoid and blocker ring locks the sun gear for overdrive and unlocks it for direct drive.

The solenoid is energized to provide overdrive above speeds of 15 mi. per hr. and de-energized at lower speeds to give conventional ratio. It also can be de-energized by a throttle-actuated switch to obtain conventional ratio at the option of the driver by pushing the accelerator pedal clear down. A relay is included in the circuit to reduce the amperage flow in the switch circuit.

Solenoid Action, Overdrive Engagement

At a speed of approximately 15 mi. per hr. the governor points "make" or complete the relay circuit, thus energizing the solenoid coil. A momentary current of 35 amperes acts on plunger *L*, pulling it into the solenoid and loading springs *M* and *N*. As this occurs, points *V* are broken, thus cutting out the main coil winding (35 amp. draw) and cutting in a "holding" coil (2½ amp. draw), sufficient to hold the plunger in the solenoid coil against the action of springs *M* and *N*. The "holding" coil is necessarily energized during the entire period of overdrive engagement. This action does not move pawl *R* attached to rod *S* because blocker ring *T* prevents pawl movement. However, spring *M* is now preloaded ready to move the pawl into engagement with the sun gear lock-up as soon as the blocker ring unlocks by torque reversal. This is accomplished by momentarily releasing the foot throttle, a necessary prerequisite for any overdrive engagement. In the diagram, the solenoid and pawl are shown in direct drive position.

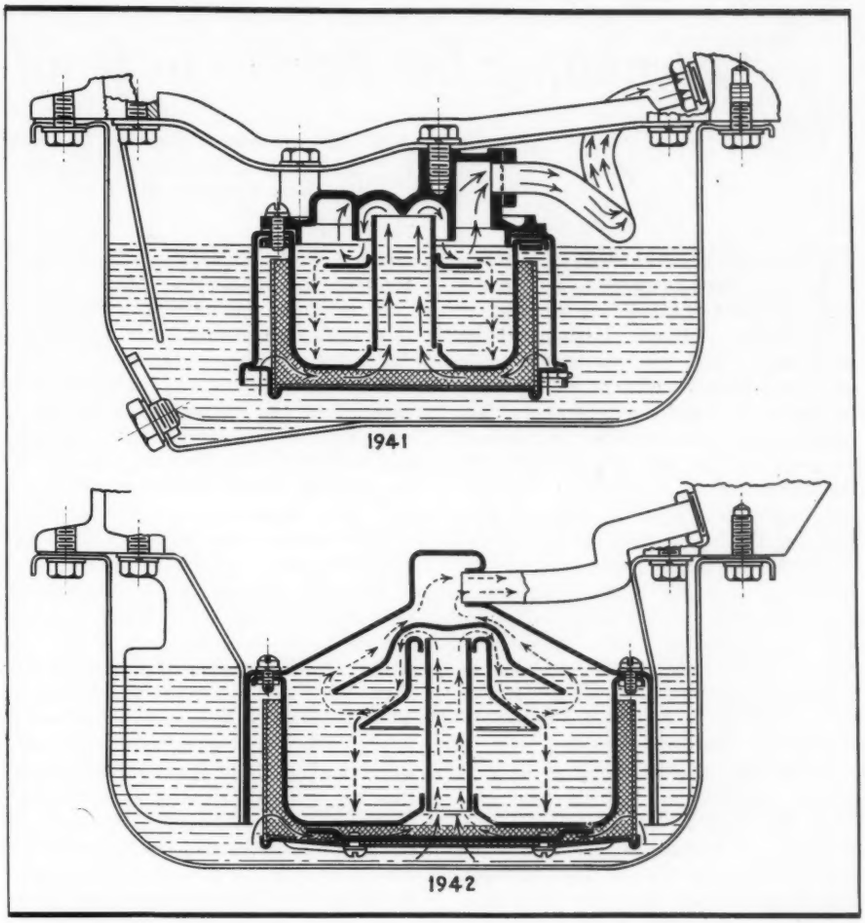
As the car coasts down to speeds below approximately 6 mi. per hr., governor points "break," thus opening the relay circuit and de-energizing the transmission control solenoid. Tension of spring *N* pulls the pawl out of the lock-up position so returning the transmission to the conventional ratio.

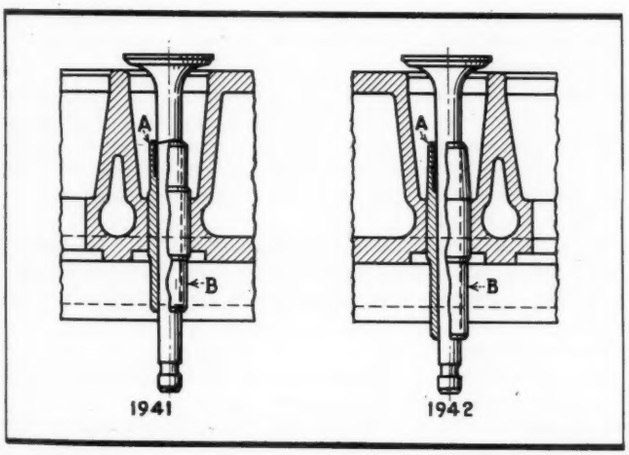
In a similar manner, as with previous overdrive transmissions, the driver can return to the conventional ratio at any time by pushing the accelerator pedal clear down (past wide open throttle position). In going past the wide open throttle position, throttle rod *K* manually contacts the plunger of the "kick-down" switch. There are two sets of contacts and two circuits through this switch. One set is in the transmission solenoid relay circuit and the other in the circuit from the ignition coil to the contact points *Y* of the transmission solenoid. The first set is normally "made" and the second set "broken." At kickdown, the reverse occurs; the first set is "broken" and the second set "made."

As the first set "breaks," the transmission solenoid circuit is broken and the transmission solenoid thus de-energized. Spring *N* is then free to pull pawl *R* out of the lock-up position. The pawl is held by engine torque until the second set of points "make." As this occurs, the ignition coil is grounded at points *Y* in the transmission solenoid; torque load is taken from pawl *R* and action of spring *N* pulls the pawl out of lock-up position, returning the transmission to conventional ratio. As rod *S* flies outward, as shown on diagram, contact points *Y* are broken and the ignition circuit restored. This action occurs in a fraction of a second, hence the driver is hardly conscious of an interruption.

New Engine Designs in 1942 Models

Improved oil cleaner design is a feature of the 1942 Pontiac. Faced with the lack of aluminum from which to make the die-cast cover of last year's cleaner, a new pressed steel design has been worked out. Dirt retaining basin is 4 times larger. A second objective of the new design was to make the oil outlet concentric with the cleaner so that each drop of oil would travel approximately the same distance through the cleaner and at practically the same velocity. This does away with eddies caused by unequal velocity of oil currents just before they reach the outlet, which might disturb dirt particles as they are settling and permit them to re-enter the oil stream. Increased efficiency of the new unit results from this concentric arrangement. The third design feature affects the character of the flow of oil as established by the contour and shape of the path through the cleaner. The new arrangement at higher speeds and even with the finer dirt is from 2 to 3 times more efficient than the one used in 1941. The 1941 cleaner took out particles of 0.00023 in. size, the new design has, in test, taken out particles one-third this size, and efficiency was found to be from twice to three times that of the original cleaner.





To reduce sticking tendency of exhaust valves caused by fuel or lubricating oil gum formations or for other reasons, a change has been made in the exhaust valve guide used on all 1942 Hudson engines. To better protect the upper portion of the exhaust valve stem from the high engine temperatures, the counterbore at the top has been decreased in diameter slightly and its length increased considerably, as shown at *A*. With this set-up, the point of contact between the valve stem and the guide at the top has been lowered approximately $\frac{3}{8}$ in. where cooling is better and heat is less intense. To compensate for the loss of bearing area at the top because of this counterbore change, the length of the guide has been increased at the bottom so that it projects $\frac{3}{8}$ in. further into the tappet compartment than heretofore, as shown at *B*. The operating clearances of 0.0025 in. for the inlet valves and 0.004 in. for the exhaust valves are continued.

A VIBRATION DAMPER has been added to the Studebaker Champion crankshaft. The camshaft is driven by a Celeron gear. Valves have mushroom lifters and horseshoe spring retainers. Lubrication is full-pressure type, with pressure to valve tappets. The gear type oil pump is mounted on the valve side.

LIGHTWEIGHT PARCO-LUBRIZED iron alloy pistons are used in 1942 Studebakers. Iron with various alloys added to give desirable characteristics is used. The new pistons are light and uniform in structure and section. Wall thickness of the Champion piston skirt is held to eight thousandths of an inch. The "Parco Lubrizing" process forms a thin protective oxide coating on the piston, which minimizes any tendency towards scuffing or scoring and produces a highly polished, long wearing surface. [Additional new design features of the cars will appear in November issue of Product Engineering.]

DETERMINING SPRINGBACK—I

Calculating Die Radius to Bend Desired Curve

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THE FABRICATION of metal structures frequently involves the bending of individual members to a prescribed contour. Tools must be built which will produce this contour with an accuracy sufficient to require a minimum of hand straightening. Customarily, the allowance for springback is estimated by the tool designer on the basis of past experience. The procedure presented herewith makes it possible for the engineering design department to calculate springback in advance, thus saving time and expense over cut-and-try methods. It is applicable to aluminum alloys, magnesium alloys, brass, alloy steels and similar metals for which the stress-strain curves do not exhibit the "knee and flat spot" peculiar to low-carbon steel.

A number of manufacturers of aluminum airplanes and buses have applied this method successfully to the forming of a variety of shapes of several aluminum alloys. In each case the predicted tool face curvature gave a final product within tolerances satisfactory to the users. The general method was developed in connection with recent investigations of roll straightening equipment and the stress-strain curves were selected as representative from a large number of such curves obtained from numerous investigations and studies conducted at the Aluminum Research Laboratories.

This method of estimating springback applies to operations where the desired deformation is produced by bending. It does not apply to forming operations which depend on shrinking, stretching or drawing of the member as a whole. Any contour which can be defined in terms of a simple circular curve or a compound circular curve can be handled. The method can be applied to a member of any shape of cross-section provided the elements of the cross-section can be determined, and provided that the original form of the cross-section is maintained during bending. The principal application of the method has been to determine the springback of aluminum alloy members formed at room temperature, but it can be extended to forming at elevated temperatures and to other metals, if the stress-strain curves for such conditions and materials are available.

In order to estimate springback by this method it is necessary to know: (1) radius of the simple curve or the radii of the compound curve which correspond to the desired final contour; (2) elements of the cross-section of the member, including the area, moment of inertia and location of the neutral axis; (3) stress-strain curve of the material for the temperature at which forming is to be done.

The tools used for the forming operation are usually wood blocks, wood blocks faced with fibre or metal, or metal blocks or dies. For closed or semi-closed shapes flexbile mandrels are usually required. In all cases the tools must fit the metal member closely and support all parts of the cross-section so that the original shape of cross-section is maintained and buckling of parts of the shape is prevented. The member is forced into contact with the block or die by hand, by a roll fastened to a lever arm pivoted at the center of curvature, by power rolls pressing the member and block into contact, or by various types of presses. The material for the tools and the equipment required for bending depend on the shape and material of the member, the stiffness of the member, and the necessary speed and quantity of production.

Two sets of formulas have been developed. One set constitutes a general method and is applicable to all shapes and amounts of forming but involves considerable computation. The second set is much simplified, but approximate. This second set of formulas may give satisfactory approximations for many forming operations. Where further precision is desirable it serves as the first step in the application of the general method.

Derivation of General Method

The general method of arriving at the curvature necessary to produce the desired final contour is to consider the member bent beyond the elastic range so that the elastic springback will be just sufficient to leave the member with that desired final contour.

In Fig. 1 is shown a typical member such as angle, which is to be formed so that the final radius of the neutral axis will be R_2. The member is bent so that the neutral axis of a short length, dS, assumes the curvature R_1 and the extreme fibres of the member are stressed beyond the elastic limit. When the bending forces are released the member tends to straighten out or "spring back" so that the neutral axis assumes a new curvature. The goal is to make R_1 such that this final radius will be R_2. It is assumed for purposes of this analysis that plane cross-sections remain plane during bending and that the entire cross-section changes angular position as indicated in Fig. 2. As the member is bent, the fibres on one side of the neutral axis are elongated and those on the other side are shortened. The amount of lengthening of a length, dS, of the extreme fibre on the tension side may be expressed as

$$e_1 \cdot dS$$

where e_1 is the total unit tensile strain, inch per inch.

Similarly, the shortening of a length, dS, of the extreme fibre on the compression side may be expressed as

$$e_2 \cdot dS$$

where e_2 is the total unit compressive strain, inch per inch. When the bending forces are released and the member springs back, the elastic recovery of the tension fibre may be expressed as

$$e'_1 \cdot dS$$

where e'_1 is the elastic recovery in tensile unit strain. And, similarly, the elastic recovery of the compression fibre may be expressed as

$$e'_2 \cdot dS$$

where e'_2 is the elastic recovery in compressive unit strain.

From Fig. 2 the corresponding angle change, Z, resulting from springback, is found to be

$$Z = \frac{e'_1 \cdot dS}{c_1} \quad \text{or} \quad Z = \frac{e'_2 \cdot dS}{c_2} \qquad (1)$$

Z = angle change from springback, radians,
c_1 and c_2 = distances from neutral axis to extreme tension and compression fibres respectively, in in.

From Fig. 1, it follows that

$$Z = X - Y = \frac{dS}{R_1} - \frac{dS}{R_2} \qquad (2)$$

Therefore, from a combination of Equations (1) and (2) it follows that

$$\frac{dS}{R_1} - \frac{dS}{R_2} = \frac{e'_1 \cdot dS}{c_1} \qquad (3)$$

or $$\frac{1}{R_1} - \frac{1}{R_2} = \frac{e'_1}{c_1} = \frac{e'_2}{c_2}$$

Since e'_1 is the elastic recovery in tensile unit strain, its magnitude may be expressed in terms of an equivalent stress and the modulus of elasticity of the material. In bending structural shapes to customary radii the biaxial stresses set up are considered negligible so that the relation between stress, strain and modulus for elastic recovery is

$$e' = \frac{f}{E} \qquad (4)$$

f = tensile stress at the extreme fibre released by springback, lb. per sq. in.
E = modulus of elasticity, lb. per sq. in.

In analyzing the forming of wide sheet and plate or the forming of solid sections bent to small radii (comparable to the section depth) biaxial stresses should be considered.

The total internal bending moment produced in the member by the bending forces must of necessity become zero when the bending forces are removed. This bending moment is that corresponding to stresses set up by the total strains compatible with the radius R_1 and, therefore, depends upon the shape of the stress-strain curve beyond the elastic limit. When this moment is released the corresponding elastic stress release may be computed by the ordinary flexure formula.

$$f = \frac{Mc}{I} \qquad (5)$$

f = tensile stress at extreme fibre released by springback, lb. per sq. in.
M = total internal moment resulting from bending forces, in.-lb.
I = moment of inertia of the cross-section, in.^{4}

Since f must be the same quantity in Equations (4) and (5), its value from Equation (5) may be substituted into Equation (4) to give

$$e' = \frac{Mc}{EI} \qquad (6)$$

The value of e' from Equation (6) for the tension fibre becomes e'_1 and may be substituted into Equation (3) to give

$$\frac{1}{R_1} - \frac{1}{R_2} = \frac{M}{EI}$$

$$\frac{1}{R_2} = \frac{1}{R_1} - \frac{M}{EI}$$

or $$\frac{1}{R_1} = \frac{1}{R_2} + \frac{M}{EI} \qquad (7)$$

This equation gives a relation between the forming radius of curvature, R_1, the final radius of curvature, R_2, and the springback in terms of the total internal bending moment M, as computed from the shape of the stress-strain curve and the forming radius R_1. The next step in the analysis then is to establish a method of computing M in terms of the total strains imposed by forming the member to the radius R_1 in the die or forming block.

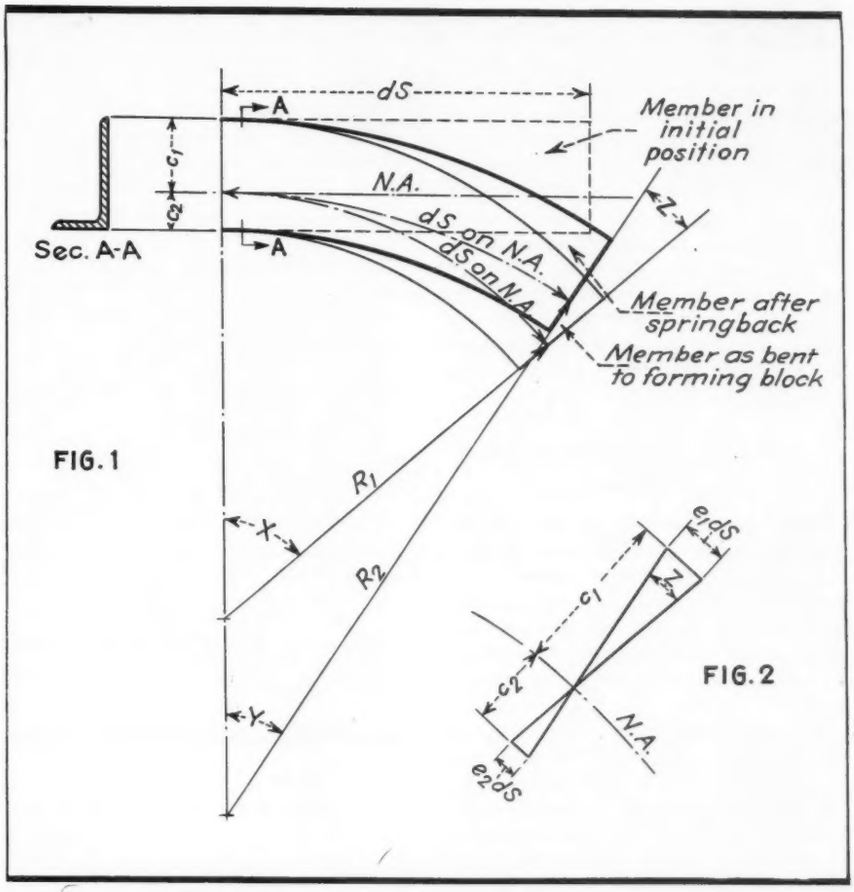


Fig. 1—Problem is to determine radius R_1 to which the member must be bent to give a final radius of curvature R_2 after springback. Fig. 2—In this analysis, it is assumed that during springback the end cross-section rotates around netural axis, as indicated

For any condition of curvature, the tensile force on a cross-section must equal the compressive force on that section and the internal moment on a cross-section must equal the external moment.

For any condition of curvature of an unsymmetrical section there will be a slight shift of the neutral axis. If the stress-strain curves are not identical in tension and compression, there will also be a shift of the neutral axis if stresses exist above the elastic limit.

In order to evaluate the magnitude of the internal bending moment, M, produced by bending, the cross-section of the member is considered as divided into appropriate elemental areas dA_1, dA_2, etc., and each area is multiplied by the stress on that area. The summation of these areas times the corresponding existing stresses, $\Sigma f \cdot dA$, for the tension side of the member must equal the summation of these products for the compression side. For a given radius of curvature R_1 the strain, e, for any elemental area at a distance, c, from the neutral axis can be determined from the relation:

$$e = \frac{c}{R} \qquad (8)$$

Knowing the strain, the stress can be taken from the stress-strain curve of the material. The location of the shifted neutral axis is determined by estimating the shift and modifying this until the condition $\Sigma f \cdot dA$ in tension equals $\Sigma f \cdot dA$ in compression. Generally two or three trials will suffice to obtain this balance.

The internal moment M, which is the sum of the moments of the forces on each elemental area about the final position of the neutral axis, may be expressed as

$$M = \Sigma\,(f \cdot dA \cdot c) \qquad (9)$$

Arrangement of this computation in tabular form as indicated in the numer-

ical examples minimizes the work of evaluating M, the internal moment.

For any assumed value R_1 the corresponding value of the resisting moment M can be computed from Equation (9) and a value of R_2 obtained from Equation (7). It is probable that the assumed value of R_1 was not exactly correct to give the desired value of R_2. If so, a new value of R_1 must be chosen and another trial made. Usually one change after the first check on the assumed value is adequate if a good approximation of R_1 can be made initially.

Derivation of Approximate Method

The determination of the proper forming radius R_1 by the above method may require several trials unless a reliable approximation of R_1 can be made as a starting point. Such a direct approximation of R_1 may, in some cases, be satisfactory without further modification.

For this purpose the actual area of the member is considered to be replaced by two hypothetical areas A_1 and A_2, the sum of which is the actual area. As indicated in Fig. 3, they are considered to be located at the extreme fibre distances c_1 and c_2, respectively from the original neutral axis.

On the basis of zero statical moment about the neutral axis, the relation between A_1 and A_2 must be such that

$$A_1c_1 = A_2c_2 \qquad (10)$$

The bending moment on such a hypothetical member may be computed directly from the total strains e_1 and e_2, corresponding to the forming radius R_1 used in Equation (8). These, in turn, may be determined as the sum of the final permanent strains, e''_1 and e''_2, see Fig. 4, corresponding to the final radius R_2 plus the elastic "springback" strains, e'_1 and e'_2, which return the stress to zero in each hypothetical area. The strains e''_1 and e''_2 corresponding to the final radius R_2 may be computed directly from Equation (8) as

$$e''_1 = \frac{c_1}{R_2}$$

and

$$e''_2 = \frac{c_2}{R_2}$$

The total strain and the corresponding stress for areas A_1 and A_2 may then be determined from the stress-strain curve by drawing lines through the strains e''_1 and e''_2 parallel to the initial modulus line to intersect the stress-strain curve. The strains and stresses corresponding to these intersections are the strains and stresses necessary to give a final permanent radius R_2. In Fig. 4 is shown a general schematic arrangement of this relation.

When these stresses are known, the value of the internal bending moment in the hypothetical member may be computed and the value substituted into Equation (7) to give the approximate magnitude of the forming radius R_1. The internal bending moment may be computed as the products of the areas A_1 and A_2 times the corresponding stresses f_1 and f_2 and their distances c_1 and c_2 from the neutral axis. Performing these steps algebraically gives

$$M = A_1f_1c_1 + A_2f_2c_2$$

But since $A_1c_1 = A_2c_2$ this equation may be reduced to

$$M = A_1c_1 (f_1 + f_2) \qquad (11)$$

The value I for the hypothetical section considered is

$$I = A_1c_1^2 + A_2c_2^2$$

Again using $A_1c_1 = A_2c_2$ it follows that

$$I = A_1c_1 (c_1 + c_2) \qquad (12)$$

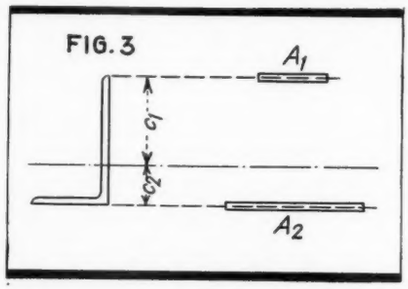


Fig. 3—Sum of hypothetical areas A_1 and A_2 equals that of cross-section and they bear relation $A_1c_1 = A_2c_2$

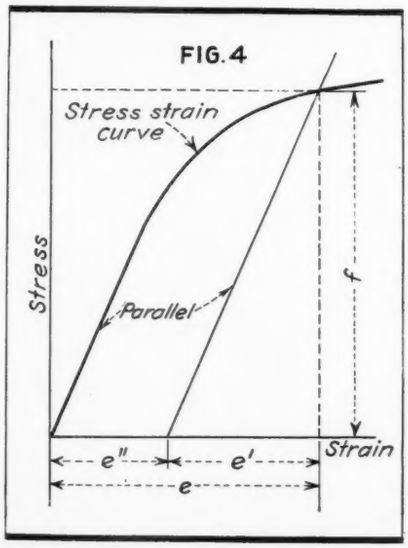


Fig. 4—Line through strain e" intersects stress-strain curve at a point corresponding to stress and strain necessary to give a final permanent radius desired

These values for M and I may be substituted into Equation (7) to give

$$\frac{1}{R_1} = \frac{1}{R_2} + \frac{A_1c_1 (f_1 + f_2)}{EA_1c_1 (c_1 + c_2)}$$

or

$$\frac{1}{R_1} = \frac{1}{R_2} + \frac{1}{E}\left(\frac{f_1 + f_2}{c_1 + c_2}\right) \qquad (13)$$

Procedure in Calculating

To determine the forming radius R_1 the following sequence of operations is suggested:

(A) Calculate the approximate forming radius R_1 from Equation (13):

$$\frac{1}{R_1} = \frac{1}{R_2} + \frac{1}{E}\left(\frac{f_1 + f_2}{c_1 + c_2}\right)$$

If experience indicates that this approximation is close enough for the particular forming to be done, no further work is necessary.

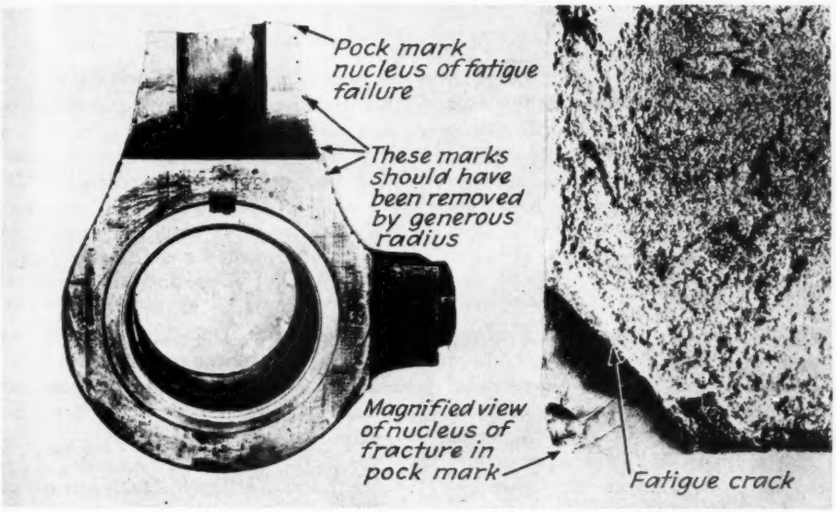
(B) If, however, this approximation is not adequate, the following steps are required:

1. Estimate the shift of the neutral axis as discussed in connection with Equation (8) using R_1 as calculated under step (A).
2. Calculate the strains on the elemental areas for the approximate curvature R_1 from Equation (8).
3. Determine the stresses corresponding to these strains using the stress-strain curve for the material in question.
4. Calculate the forces $\Sigma f \cdot dA$ for tension and compression on the cross-section. These forces should be approximately equal. If not equal, assume a different shift of axis and recalculate until satisfactory equality is obtained.
5. Determine the internal moment from the summation for the elemental areas from Equation (9).
6. Using this internal moment, calculate R_1 from Equation (7) and R_2.
7. This value of R_1 should agree with the assumed value of R_1. If it does not, using the value of R_1 from step 6 above, repeat the procedure until satisfactory agreement is obtained between the assumed and calculated values of R_1.

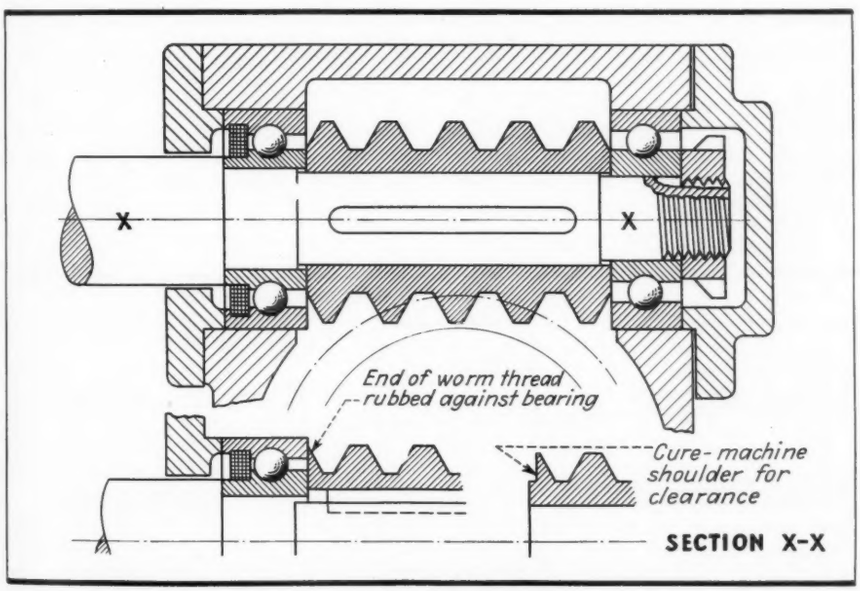
Obviously these calculations cannot be made unless the stress-strain curve for aluminum alloys is available. It is necessary to refer to the stress-strain curve of the particular alloy of which the part is made.

Stress-strain curves for the aluminum alloys in most common use and an example of the application of this method will be given in Part II of this article which will appear in the November number. The die radius of the forming block for bending a channel-shaped member to a specified radius will be calculated.

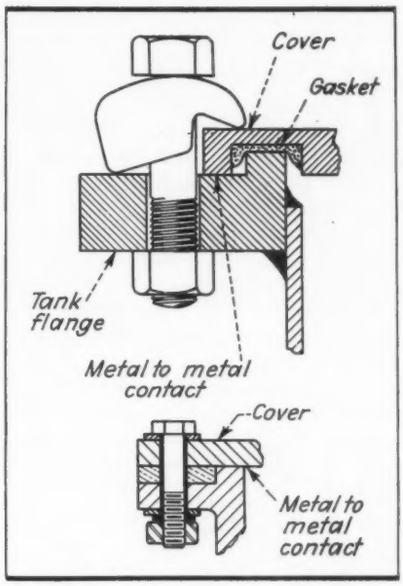
Causes and Cures


Pock mark nucleus of fatigue failure
These marks should have been removed by generous radius
Magnified view of nucleus of fracture in pock mark
Fatigue crack

Pock marks made by a prick punch while laying out this locomotive side rod caused its failure in service. These sharp nicks were excellent starting-places for the fatigue cracks, and should have been removed by grinding a radius. A generous radius should be specified for all sharp machined edges of parts such as this. It is not enough to chamfer; that simply leaves two sharp edges instead of one, particularly dangerous if the chamfering is done roughly. Radius should be smooth, even polished.



Worm drive unit of a new automatic machine overheated during its trial run. The design was checked again and found correct for lead and speed, but the designer had overlooked the fact that the standard commercial worm, which he had selected from a manufacturer's catalog, had no hub on its ends. At the two points where the worm thread ran out to the ends it rubbed against the outer races of the ball bearings. Cure was to machine a small shoulder on the ends of the worms for clearance. The job was not complete until a detailed drawing specifying this machining had been prepared.



Cork gaskets lose their effectiveness if compressed beyond their elastic limit. Effectiveness of many types of joints depends entirely on the correct amount of tension being put on the bolts. If too much pressure is applied, the gaskets can be crushed so that they no longer seal the joint. Allis-Chalmers have developed the metal-to-metal joints shown above for power transformers. These joints compress the gasket a predetermined amount.

WHITE INSTRUMENT DIALS coated with lacquer or paint are often discolored by high temperatures, light, chemical fumes and moisture. Westinghouse research engineers have recently developed a liquid plastic material that is sprayed on specially prepared sheet metal under controlled atmospheric conditions to produce a hard, smooth, white surface that is unaffected by these conditions. Lithographing the dial scales coated with this smooth plastic also presented a problem. In damp atmospheres the plastic is practically impervious to ink. Controlled dry atmosphere of the printing room was the cure.

PRODUCT ENGINEERING will pay a minimum of $3 for each example published in Causes and Cures. Where illustrations are necessary, include drawings, rough sketches or photographs.

SHORTER SLEEVE BEARINGS

Designed for Widest Range of Conditions

E. R. SUMMERS

Induction Motor Engineering Department, General Electric Company

THE CONVENTIONAL sleeve bearing with steel shell, hard babbitt lining and straight horizontal oil grooves, has been used successfully for many years. However, this dependable type of bearing has occasionally been challenged by the trend in industry toward flexible, individual motor drives instead of the old, cumbersome central drives with complicated line shafting arrangements. These newer individual chain, flat belt, or V-belt drives commonly have a short distance between pulley centers to conserve belt material and floor space, a low belt speed to reduce the size and cost of the larger driven pulley, and an unfavorably small arc of belt contact on the motor pulley. Also some form of pivoted motor base is often used to maintain automatically the belt tension at a predetermined amount, which is frequently the highest value that the motor manufacturer will approve for the bearings. These factors have tended to increase the duty imposed on motor bearings, and the load limitations of conventional sleeve bearings have sometimes been exceeded.

To meet the challenge of modern trends in belted applications, the fundamentals of sleeve bearing design have been critically reviewed. This article presents the results of extensive tests that have been made both in the factory and in the field under a wide range of service conditions to determine the most advantageous ratio of bearing length to shaft diameter and to investigate different oil grooving arrangements, in order that sleeve bearings could be better designed for a maximum of reliable service with the most effective use of active material.

Pounds Per Square Inch Fallacy

In the design and application of sleeve bearings, it has been a common practice to express the load in terms of unit pressure or "pounds per square inch" of projected area. The often accepted, yet fallacious axiom, that bearing capacity is proportional only to projected area, bearing length times diameter, has become so deeply implanted that many persons appraise a bearing solely by its size and weight.

The early motors of 40 years ago usually had long bearings, with a length at least three times the shaft diameter. Later a shorter length of 2½ diameters was considered desirable, and more recently lengths of 2 and 1½ diameters or less have been advantageously used. In occasional comparative tests it has been observed that short bearings would stand as much, and sometimes more, belt tension as longer bearings of the same diameter. Obviously, there must be some other factors in bearing design that are of equal or greater importance than size.

Actually the length to diameter ratio of a bearing, the arrangement of oil grooves, or the direction of rotation are often far more significant than the amount of projected area in determining load capacity. It is possible to make bearings too large as well as to have them too small. Data are presented in this article on specific cases where the bearings were found to be too long or to have too large a diameter to give the best possible performance.

Effects of Shaft Deflection

Field examinations of long bearings which had been subjected to heavy horizontal belt tension showed that the bore was no longer uniform after operation for a short time at excessive temperatures, or for a longer period at lower temperatures, but was enlarged at the end adjacent to the belt pulley, as shown in Fig. 1(a). This enlargement occurred only in the horizontal axis of the belt pull, there being no corresponding increase in the vertical diameter. The belt tension had deflected the shaft so that the load was concentrated initially on about 30 percent of the bearing surface at the end portion between *A* and *B*, Fig. 1(b), and in extreme

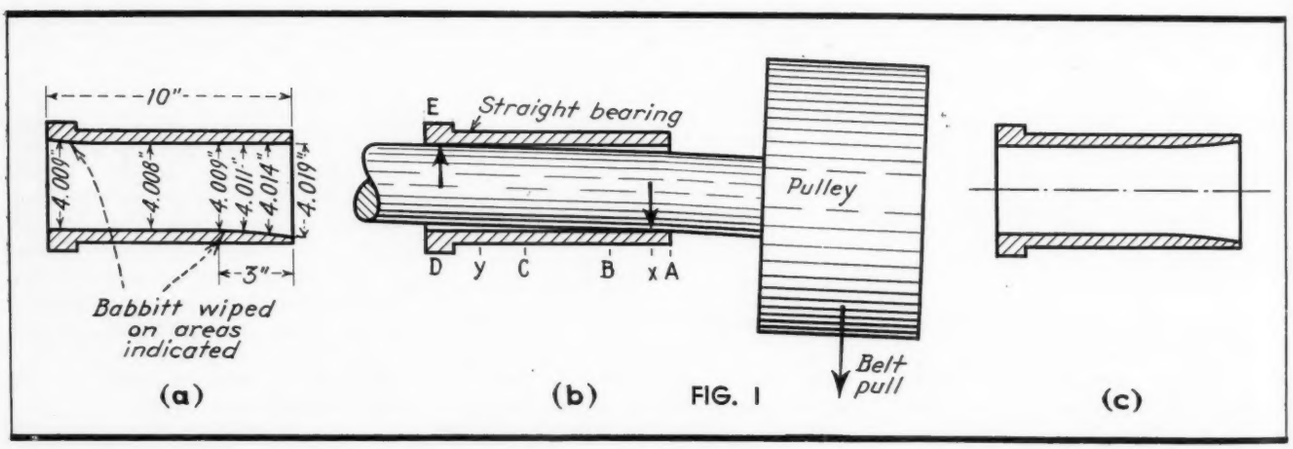


Fig. 1—Effects of shaft deflection on long bearings—(a) gives the micrometer measurements of the bore of a worn bearing across diameters parallel to the line of resultant belt pull; (b) effects of shaft deflection on load distribution over the length of a long bearing; (c) long bearing bored to a curve corresponding to the shaft deflection so as to distribute bearing pressure uniformly

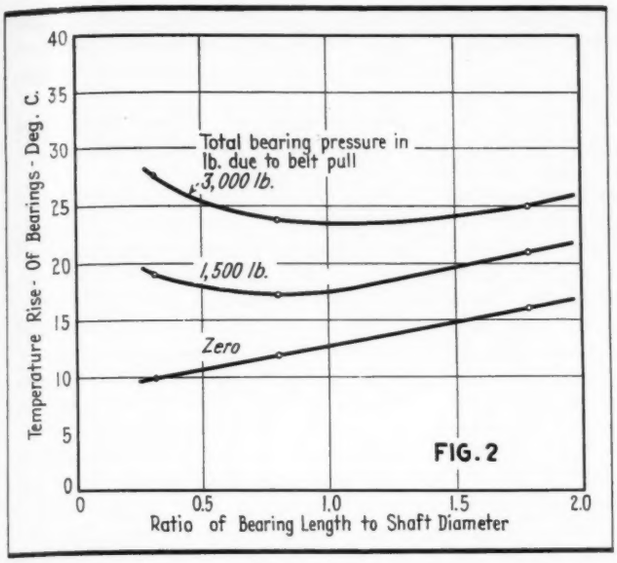


Fig. 2—Belted tests on 2.75 in. dia. sleeve bearings of various lengths with shaft running at 1,800 r.p.m., showing temperature rise of bearings under load and no-load conditions

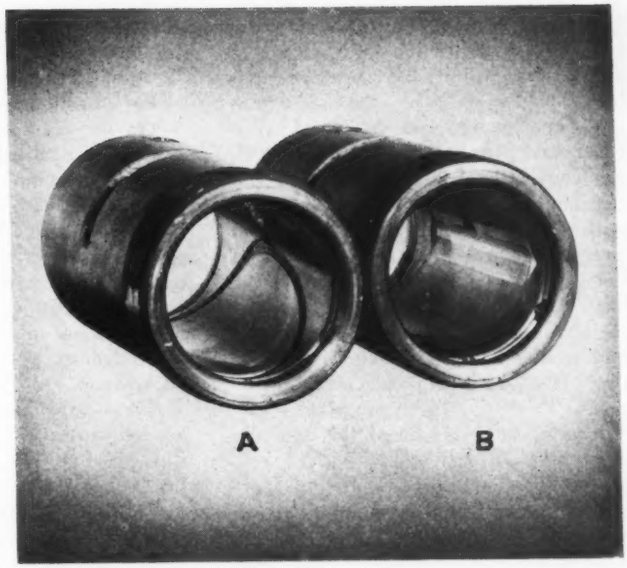


Fig. 3—Comparison of oil groove patterns. At (A) is shown a bearing with spiral grooves, and at (B) is shown one with conventional straight groove parallel to the axis

cases pressure existed at the back end on the opposite side at *E*. For initial operation with belt tension, the temperature of the bearing shell at point *x* was usually higher than at point *y*, as much as 20 deg. C. higher in extreme cases.

When the babbitt of a long bearing starts to wipe at *A*, the wiping progresses toward *B*, *C*, and in extreme cases to *D*, thereby distributing the load over a greater area and relieving any pressure on the opposite side at *E*. Sometimes a bearing will wipe slightly over only part of its length, cool off, and then continue to run indefinitely without the motor user being aware that one end of the bearing overheated during the first few minutes of operation. With careful control of belt tension, a long bearing can be "broken in" by operating at a temperature which will cause the babbitt to flow slowly and conform to shaft curvature, thereby increasing the pressure area until the load can be carried with a reasonable bearing temperature.

However, bearings must be rated in terms of initial capacity if a "breaking in" period is to be avoided. The greater part of the bearing length in Fig. 1(b) is not only useless for initial operation, but is actually detrimental, because any pressure on the back side at *E* must, by summation of vector forces, increase the pressure at *A* above the value which would have existed with a shorter bearing. To utilize initially the full surface of long bearings, without first wiping babbitt, it would be necessary to make them with a curved bore as in Fig. 1(c) corresponding to the shaft deflection. To try this out, several new bearings were made with various curvatures of the bore. The test results, although rather favorable, did not indicate this to be a completely practical method of utilizing long bearings.

Bearings of Different Lengths

Long bearings are sensitive to alignment in assembly, to deflection of motor endshields and frames under stress, or to a small taper unintentionally machined into the bore of the bearing. Load is not carried on those portions of a bearing having a clearance with the shaft greater than the thickness of an oil film under pressure. Some erratic test results with long bearings were traced to small variations in alignment and taper which were within reasonable tolerances of manufacture and assembly.

In a series of belted tests on an 1,800 r.p.m. motor with 2¾ in. diameter journals, the bearing length was reduced progressively to determine the maximum surface that could be effectively used without sacrificing either load capacity or temperature rise.

With the belt removed and motor running light, the minimum temperature rise occurred with a bearing length of less than one inch, or a length to diameter ratio of less than ½. This "zero load" curve of Fig. 2 represents a direct connected motor with good alignment. With 1,500 lb. bearing pressure, corresponding to normal load belt pull, the lowest temperature was obtained with a length equal to 0.8 of the diameter. At a heavy overload of 3,000 lb. a length of about 1.1 diameters would give the coolest operation.

As a result of these tests and other factors discussed later, the conclusion can be drawn that a bearing length of 1½ diameters provides the greatest surface that can be effectively used. even for possible abnormal conditions. A length of 1½ diameters is considerably less than has heretofore been used on general purpose motors. This shortening of the bearings results in less friction loss, lower temperatures, smaller housings, less obstruction to the flow of ventilating air into the motor, and less susceptibility to deflection or slight misalignment. These shorter bearings have been incorporated as standard in the General Electric Tri-Clad motor.

Importance of Oil Grooving

There has been a long-standing concept that sleeve bearings lubricated only by oil rings inherently have much less load carrying capacity than pressure lubricated bearings, where oil is forced in through a pipe. Tests have shown, however, that the load capacity depends not on the method of delivering oil to the bearing, but rather on how the lubricant is distributed within the bearing by the oil grooves. With pressure lubrication the supply of oil is sufficient to fill continuously the clearance between bearing and shaft, but if this clearance can be filled adequately from oil rings, the bearing will carry just as much load. Recognizing these facts, changes have been made in the grooving to provide a "full" bearing, and thereby more effectively utilize the inherently good load capacity of ring lubricated bearings.

Straight, "dead-end," horizontal oil grooves as shown in Fig. 3(B) have

long been used in conventional sleeve bearing design. Although satisfactory for many applications, these bearings have "dead-spots" where the load capacity is sharply reduced. At Fig. 4(a) is illustrated the condition under which the conventional bearing is most vulnerable, namely, horizontal belt pull with tight side underneath. Oil is delivered from the rings to the groove on the loaded side, but the pressure of the shaft against the groove prevents oil from entering and filling the clearance, and overheating may occur at a relatively low load from insufficient lubrication.

In Fig. 4(b) is shown the opposite shaft rotation with tight side of belt on top. The conventional bearing operates more satisfactorily under this condition because oil deposited in the groove on the "clearance" side can enter the bearing freely, wet the lower portion of the shaft, and form an oil film "wedge" as the center of pressure is approached.

Tests showed that the conventional bearings operated cooler and more consistently under condition (b) and had 2 to 4 times as much load capacity as under condition (a). For horizontal belt pull, condition (a) is not only more common, but is required on most pivoted base applications where the belt tension increases automatically with the load. The conventional bearing therefore has its least load capacity and is most erratic in performance for the very condition under which it is most frequently used with belt or chain drives.

For the limiting condition of Fig. 4(a), the load capacity could be increased by changing the oil groove location from *o-c* to *o-d* so that it would not coincide with the center of pressure on the bearing. Although tests with oil grooves at position *o-d* were more favorable for a particular angle of belt pull, this shifting of "dead-spots" from one location to another did not offer a satisfactory general solution—critical "dead-spots" must be eliminated.

After many tests with different oil groove patterns, a type of spiral groove illustrated at (A) in Fig. 3 was adopted for general purpose motors. There are no critical "dead-spots" with these spiral grooves, because only a small percentage of the babbitt surface is removed along any axial line. Fig. 4(c) shows the superior lubrication obtained with the new design. The shape and "continuity" of the spiral grooves, the pumping action of the rotating shaft surface, and the low-pressure area in the clearance beyond the center of pressure cause the oil to be sucked into the grooves freely and distributed quickly throughout the bearing for all conditions of operation thereby achieving the fundamental objective—a "full" bearing. This basic improvement in lubrication gives good load capacity for belt pull in any direction, has more nearly uniform characteristics for opposite directions of rotation, assures longer life, and provides a greater factor of safety against interruptions of service.

Advantages of Short Bearings With Spiral Grooves

The vertical line of Fig. 5 illustrates the erratic results obtained with conventional long bearings with straight oil grooves and operated under the most unfavorable condition as shown at (a) in Fig. 4, with tight side of horizontal belt underneath. Seven similar bearings were tested in different motors at the same load of 2,250 lb. Each point on the vertical line represents the ultimate temperature rise for continuous operation of a specific bearing. The temperature rises of five bearings ranged from 26 deg. C. to 120 deg. C., and two bearings failed before steady conditions were reached. When shaft pressure against the groove blocks the free entrance of oil, the performance becomes erratic because quite small variations in alignment, shaft deflection, taper, condition of shaft surface and bearing bore, edges of oil grooves, etc., will cause large differences in initial load capacity and temperature rise.

The horizontal line of Fig. 5 represents the average of five different tests with bearings only half the length of conventional bearings and having spiral oil grooves as shown at (A) in Fig. 3. The highest temperature rise of 23 deg. C. at 5,200 lb. load, 260 lb.

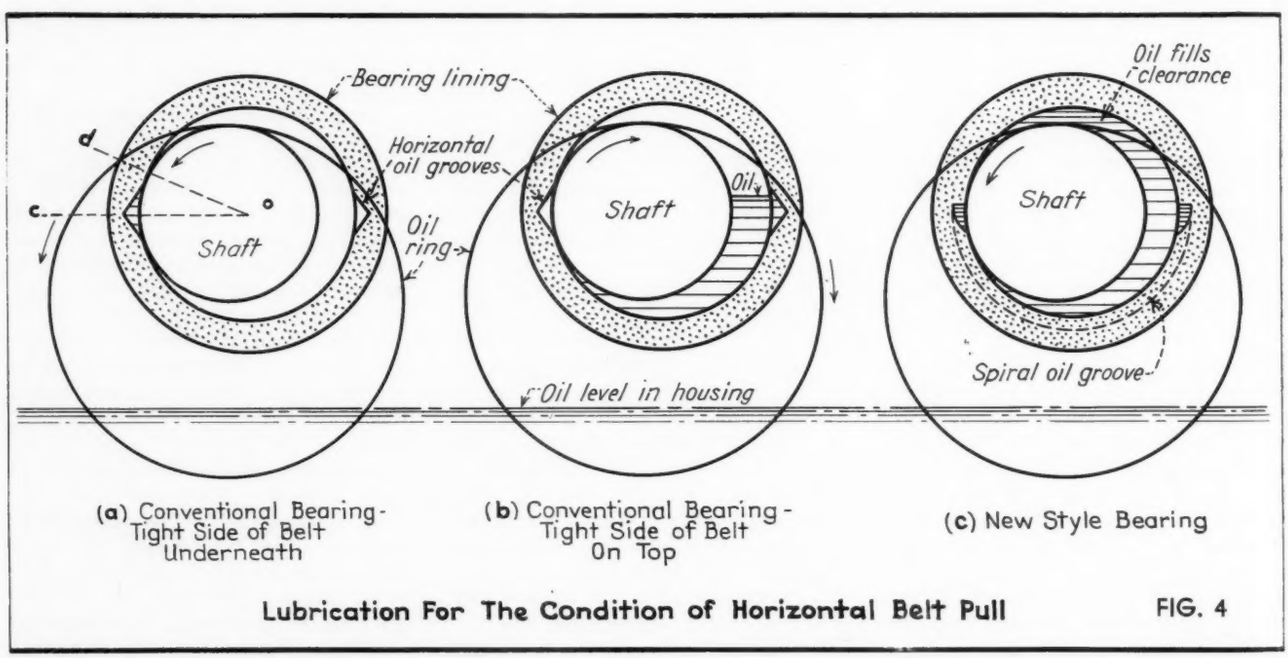


Fig. 4—At (a) is shown the condition created with a horizontal belt pull with tight side underneath. The oil is delivered to the groove on the loaded side. With the tight side of the belt on top, as at (b), conditions are better because the oil is delivered to the groove on the clearance side. At (c) is a new-style bearing with spiral oil grooves which give improved lubrication for any direction of belt pull and either direction of rotation

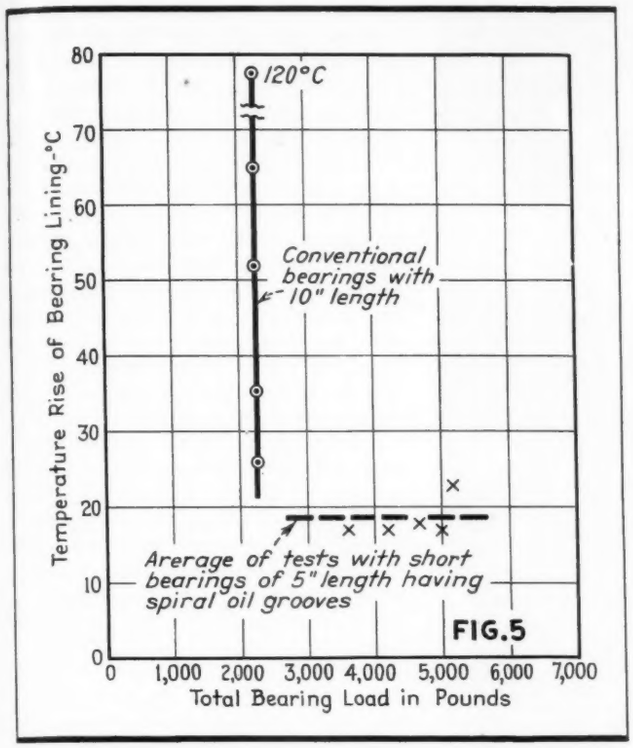


Fig. 5—Belted tests on two 4 in. diameter sleeve bearings, one short and the other one long, with journals running at 720 r.p.m.

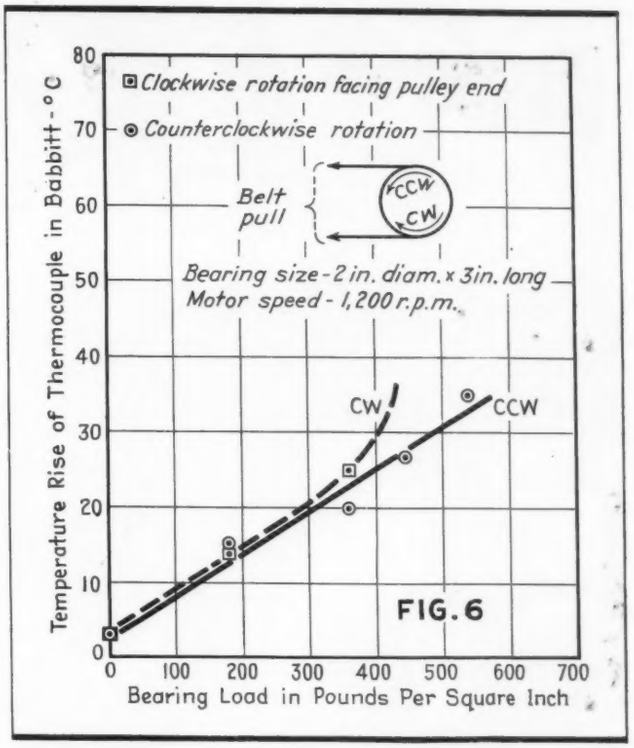


Fig. 6—Comparison of temperature rise in new style short bearings when rotating clockwise and counter-clockwise

per sq. in., with the short bearings is 3 deg. C. better than the most favorable test of 26 deg. C. rise at 2,250 lb. load. 56 lb. per sq.in., with the conventional bearings. All of the tests shown in Fig. 5 were taken with the same angle of belt pull and mechanical rotation.

Obviously, sleeve bearings cannot be designed and applied on the basis of "pounds per square inch" only. Since the combined advantages of better lubrication and shorter length make the new style bearings more consistent in performance and less susceptible to variations in manufacture, assembly, or application, a general review can be made of sleeve bearing limitations.

A broad outline has been presented of the major steps taken in arriving at this improved bearing design. The following information not only recites more completely many field tests and studies which substantiate the conclusions made, but also discusses additional principles of good bearing design that should be of general interest and value to machine designers.

Limitations of Short Bearings with Spiral Grooves

A new style bearing 2 in. in diameter and 3 in. long was tested at 1,200 r.p.m. with progressive increases in belt tension to investigate load limitations. Each point on the curves of Fig. 6 represents a steady state condition with constant bearing temperatures. For counter-clockwise rotation, tight side of belt underneath, stable operation at 35 deg. C. rise was obtained at a pressure of 540 lb. per sq. in. of projected area, but the test was discontinued at that point because of excessive bending stresses in the motor shaft. For clockwise rotation the bearing temperature became unstable at pressures above 400 lb. per sq. in. Results very similar to those of Fig. 6 were also obtained at 1,800 r.p.m.

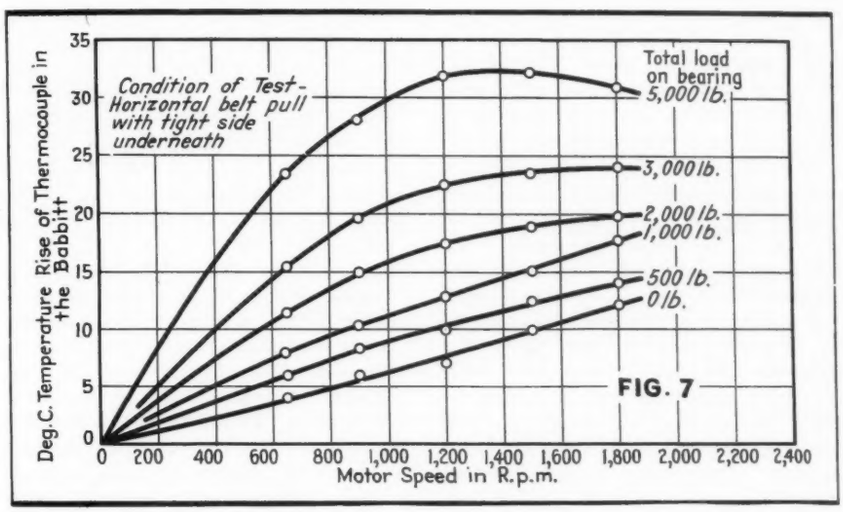


Fig. 7—Belted tests on bearings 2¾ in. dia. and 2¾ in. long with spiral grooves

with a bearing 1½ in. in diameter and 2¼ in. long.

In Fig. 7 are shown test results for a 2¾ in. x 2¾ in. bearing with spiral grooves over a load range from zero to 5,000 lb. at speeds from 600 to 1,800 r.p.m. with tight side of horizontal belt underneath. Each plotted point represents a period of operation until bearing temperatures became constant. The 5,000 lb. load corresponds to a bearing pressure of 660 lb. per sq. in. The "zero" curve refers to zero external

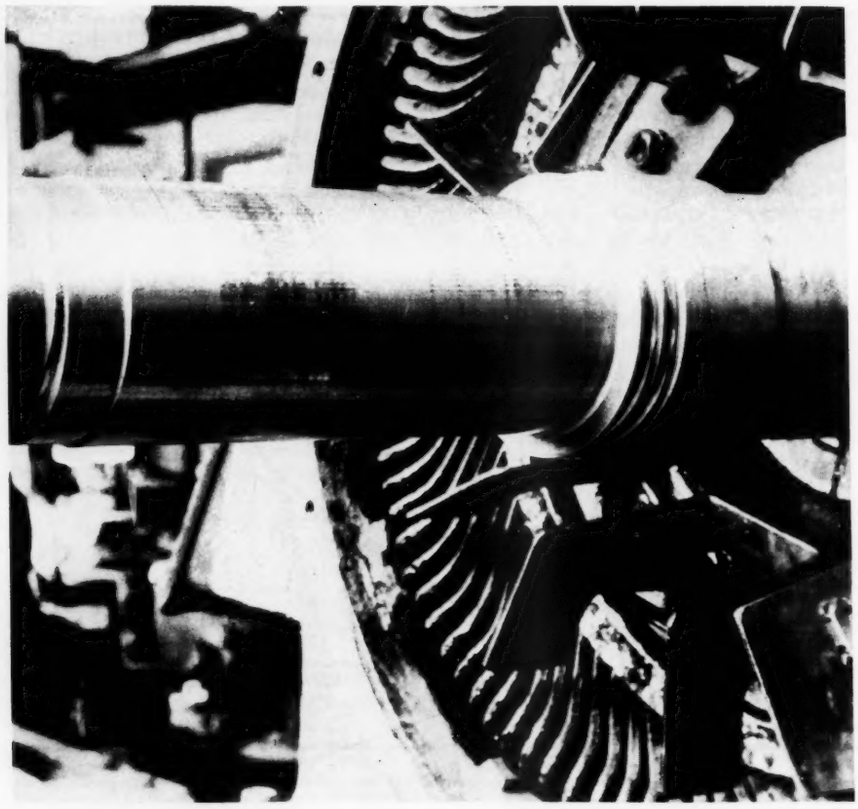


Fig. 8—Ground journal of a new shaft after one hour of running in a 4x8 in. bearing at 155 lb. per sq. in. pressure. Chatter marks from grinder are visible

load, that is, belt removed, the weight of motor rotor being neglected because it corresponds to only 15 lb. per sq.in. unit pressure. Temperatures were measured with thermocouples embedded directly in the babbitt lining. Later when the bearing was operating at 740 lb. per sq. in., the motor shaft broke from bending stress and the tests were discontinued.

These tests at extremely high pressures were made to investigate the inherent load capacity of babbitt with oil ring lubrication and to obtain a better idea of available factors of safety in the design and application of sleeve bearings. There was no intention of using such pressures in design, because shaft stresses, belt tension limits, pulley construction, and other practical considerations usually restrict the bearing pressures to less than 200 lb. per sq. in. even with short bearings having a length of only 1½ in. diameters.

There is also no intention to infer that general purpose short bearings with spiral grooves may be used satisfactorily for all conditions of operation at the extremely high pressures mentioned above. The initial load capacity with a new bearing and freshly ground shaft is less than the ultimate capacity with progressive increments in pressure after the shaft has acquired a polish and any high spots on the babbitt have been worn off. Fig. 8 shows a new shaft after one hour of initial operation with a 4-in. x 8-in. bearing at 155 lb. per sq. in. pressure. Shallow chatter marks from the grinder are visible and there is some discoloration of shaft surface by the lubricant although the initial temperature rise of babbitt did not exceed 35 deg. C. After operating one day at 190 lb. per sq. in., this shaft surface acquired a uniform polish, chatter marks were no longer visible and the bearing temperature rise decreased to 24 deg. C. There was no observable change in either shaft or bearing diameter by micrometer measurement.

Motors are sometimes started under belt tension with dry bearings after long shutdown periods. Also in cold weather the oil may be so heavy that good lubrication is not established immediately after starting. There are many other variables such as vibration, different grades of oil, changes in ambient temperature, alignment, dirt in oil, and shock loads, which affect bearing performance. There must be a considerable margin between laboratory tests and field applications to provide an adequate factor of safety against those hazards of service which cannot be well anticipated.

The new type bearings have been tested under pressures commonly considered impossible for oil ring lubrication, and have performed successfully on applications where conventional bearings have failed. Although the new bearings are free of critical "dead-spots" and have adequate capacity for any normal operating condition, there is no intention to imply that a given general purpose bearing such as that shown at (A) in Fig. 3 is entirely symmetrical in its performance. Whenever the smooth surface of a bearing bore is interrupted with an oil groove, the load capacity is reduced to some extent for one or more operating conditions. High oil film pressure is not maintained at or immediately adjacent to a groove. Since no way has been found to avoid oil grooves altogether, the next best thing is to use the simplest, continuously curved groove which will circulate oil to both ends of the bearing and to both sides of the shaft, and still not interrupt the full length of bearing surface in any axis.

Oil Groove Patterns

When the straight "dead-end" oil grooves of the conventional bearings were shifted 20 deg. above the center of pressure, from position *o-c* to *o-d* in Fig. 9, the load capacity for horizontal belt pull to the left was improved for counter-clockwise rotation. But successive tests were inconsistent because the groove was yet too close to provide adequate lubrication and load distribution as the center of pressure was approached. The bearing still had much more load capacity for clockwise than for counter-clockwise rotation, however, this dissymmetry can be readily explained by contrasting the large 200-deg. arc of smooth continuous bearing surface between groove G_2 and the center of pressure with the small available 20 deg. arc below groove G_1 as shown in the figure.

Any interruption in bearing surface as the center of pressure is being approached greatly decreases the load capacity, but a similar irregularity on the trailing side is of much less consequence. This is explained by the fact that the oil film pressure builds up gradually to a peak value and then decreases quite suddenly after the minimum clearance is passed, because the excess oil is squeezed out at this point. It is not sufficient to say that an oil groove must be displaced from the center of pressure by a specified angle of so many degrees for good load capacity. The direction of rotation must be considered in any particular case.

This same basic principle explains the smaller amount of dissymmetry in a spiral grooved bearing. In Fig. 10 at

(a) is shown the bottom half of a bearing with a groove pattern well adapted to general purpose motors. For counter-clockwise rotation with horizontal belt pull to the left, the center of pressure occurs at *x-x*, oil is delivered to the groove at *a*, is sucked and pumped under shaft to low pressure side *b* where it leaves the groove, fills the clearance indicated at (c) in Fig. 4, and is carried without interruption over top of the shaft toward the pressure area, thereby providing maximum load capacity.

For clockwise rotation the oil enters at *b* and passes under the shaft, but that portion of the groove near *a* must be passed over as the center of pressure at *x-x* is approached, thereby reducing the load capacity of the middle part of the bearing. But the end portions are far enough away from the grooves to carry a higher pressure.

As the center of pressure is shifted from the side of the bearing at *x-x* to the bottom at *y-y*, the reduction in load capacity of the end portions is accompanied by an increase in the capacity of the middle part since the grooves spiral out toward ends of bearing at the bottom. Extensive tests with different angles of belt pull have shown that for any operating condition, a sufficient part of the bearing length remains effectively lubricated to carry the loads commonly imposed on general purpose motors.

On large direct connected motors beyond the belted range where the center of pressure occurs at axis *z-z* on bottom portion near vertical centerline, the "continuous" rectangular oil groove shown at (b) in Fig. 10 will give higher load capacity near ends of bearings than the groove shown at (a) in Fig. 10. With this rectangular pattern there is an arc of at least 60 deg. between the groove and center of pressure throughout the effective length between points *P* and *Q*, and this insures high capacity and symmetrical performance for both directions of rotation. These two patterns, and other similar arrangements, have been used successfully with pressure lubrication, with oil ring lubrication, and with various combinations of the two. For any uncommon, special or particular situation, with a little ingenuity, a tailor-made pattern can be devised by following the basic principles of a "full" bearing and by not interrupting the surface over an arc of about 60 deg. on the leading and 20 deg. on the trailing side of the center of pressure.

With the "continuous" type of groove, the oil circulation cannot be blocked with air pockets as sometimes occurs in "dead-end" grooves. Fig. 11 represents a wide range of experience with "continuous" oil grooves. The smaller oil ring bearing, size 1½ in. x 2¼ in., was applied on a 3-hp. textile motor and the large pressure lubricated bearing, size 19 in. x 24 in., was used on a 15,000-hp. motor.

Limitations of Split Bearings

The tests so far described were made with solid steel tube bearings such as illustrated in Fig. 3. An extensive investigation was also made of split bearings with diameters of 2 to 5½ in. Split bearings commonly have the joint on the horizontal center line as shown in Fig. 12. For horizontal belt pull the center of bearing pressure coincides with the joint, and the oil film is usually not fully maintained because lubricant will be forced out of any opening. If the joint surfaces are carefully made, and the two halves of bearing firmly held together without distortion, the performance approaches that of a solid bearing. If such precautions are not taken, the load capacity in the horizontal axis may be sharply reduced, by 50

Fig. 11—Continuous oil grooves have been used on a wide range of bearings. The small bearing is for a 3 hp. motor, the large one for a 15,000 hp. motor

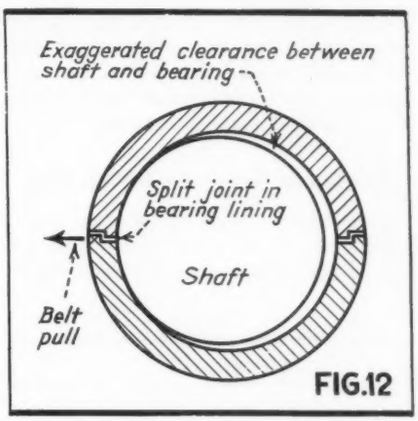


Fig. 12—Split type of sleeve bearings commonly have the joint on the horizontal center line. Horizontal belt pull causes the pressure to come against the split

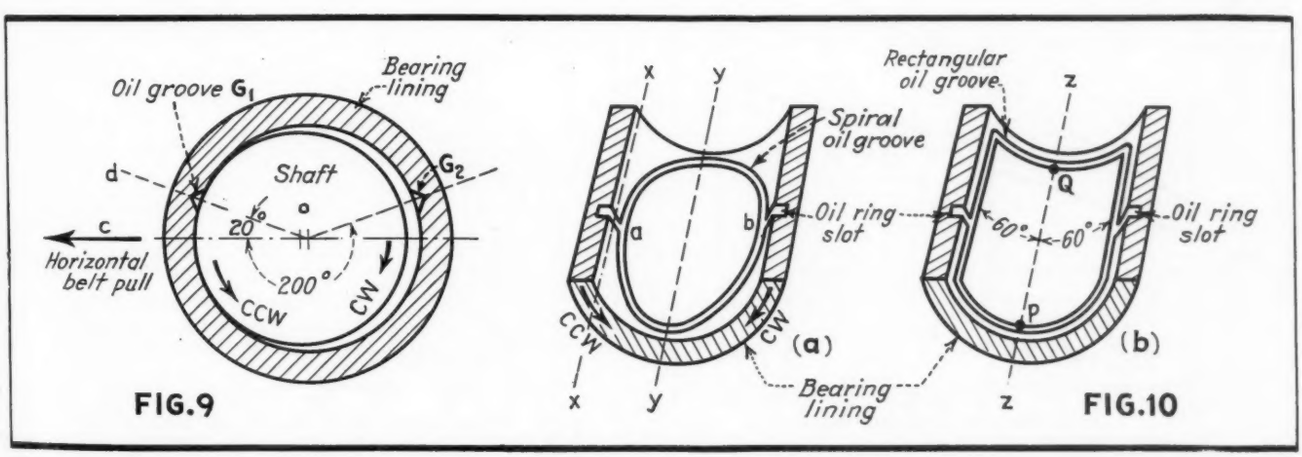


Fig. 9—With the conventional horizontal "dead-end" oil groove, the oil flow may become blocked. Fig. 10—Typical "continuous" oil groove patterns. The design shown at (a) is for general purpose motors. Design (b) is for large direct-connected motors

percent or more. The load capacity in the vertical axis is not greatly influenced by small variations in split joints. Most motors with split endshields are directly connected to the load, instead of belted, for which case the bearing joints are of less consequence.

Bearings of Different Diameters

Comparative belted load tests were made on a 900 r.p.m. motor using first a 4 in. and later a 5 in. diameter journal on the pulley end as shown by the two curves of Fig. 13. All conditions of test were as nearly identical as possible except for the diameters of bearing and shaft. Because of the higher rubbing speed of the shaft journal, the larger bearing had a greater temperature rise than the smaller bearing, 23 percent more rise at zero external load and 50 percent more at 7,000 lb. load. The temperature rise under field service conditions increased approximately as the square of the bearing diameter.

These tests showed that bearings may be too large as well as too small. Bearings should not be appraised on the basis of their size and weight. In some cases better performance can be obtained with a smaller bearing.

Effects of Different Lubricants

Comparative belted tests were made on a 1,200 r.p.m. motor having 2 in. x 3 in. bearings with four different lubricating oils designated as *A*, *B*, *C*, and *D* in Fig. 14. Each point on the curves represents a steady state condition after several hours of operation. All tests were made consecutively under the same conditions, the motor being stopped only long enough to drain and refill the bearing housing with a different grade of oil. The heaviest oil *A* was labelled SAE-40. The lightest oil *D* was in the SAE 10 range and had a viscosity of about 200 sec. Saybolt at 100 deg. F. The other two intermediate oils *B* and *C* were labelled SAE 30 and SAE 20. With belt removed and motor running light, corresponding to zero load in Fig. 14, the bearing temperature rise was 3 deg. C. with the lightest oil *D* and 17 deg. C., or almost six times as much, with the heaviest oil *A*. At the extreme load of 500 lb. per sq. in. the temperature rise was approximately 35 deg. C. and was about the same for all four oils. At 100 lb. per sq. in., which is in the normal operating range, the bearing temperature rise with the heaviest oil was more than twice that obtained with the lightest oil.

Some persons believe that a heavy oil is required for extremely high bearing pressure, but these test results indicate equally good performance with the light oil. The motor used for the tests of Fig. 14 was operated continuously for 30 days, 24 hr. a day, at a bearing pressure of 500 lb. per sq. in. with the lightest oil *D*, and there was no observable wear of bearing or shaft by micrometer measurement. Field measurements also indicated no wear on two different 4-in. diameter new type bearings after more than a year of continuous operation under heavy belted loads with light oil.

Since oil is usually circulated rapidly through bearings with spiral grooves, it is desirable to deliver an adequate amount of oil with the rings. When testing some of the larger bearings at high pressures it was found advantageous to use two oil rings side by side in the same slot in order to keep the bearing clearance continuously filled.

Conclusions

This review of the motor sleeve bearing situation indicates that marked improvements are still possible in the most familiar and widely used devices, regardless of how long they have been under development. It also indicates that sleeve bearings are well able to carry loads far beyond the usually accepted values, provided that the bearing dimensions are proportioned for correct distribution of load, the bearings are properly grooved, and there is an adequate supply of oil to the high pressure area.

In November PRODUCT ENGINEERING, the author will explain in detail the test procedure used in the measurement of the bearing loads and their adjustment to the value desired.

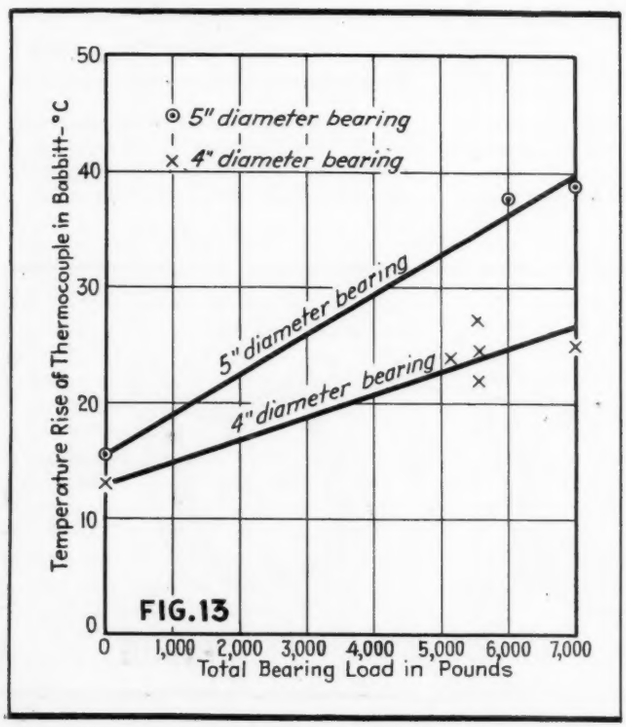


Fig. 13—Belted tests on bearings of different diameter

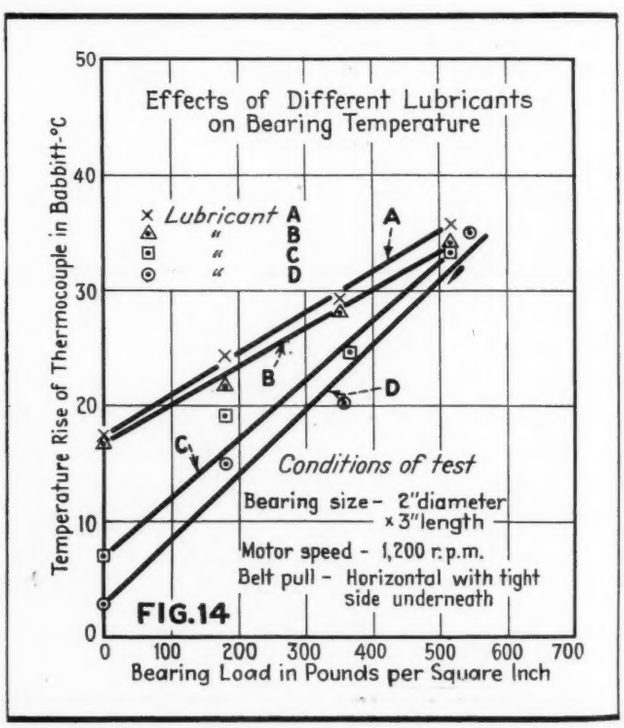


Fig. 14—Effects of different lubricants on bearing temperature

DESIGN OF HOOKS

In Bending and Direct Stress

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LET THE HOOK shown in Fig. 1 (*a*) be loaded with the force P which passes through the center of curvature of cross-section BC. As shown in most books on strength of materials (for example, "Strength of Materials" by S. Timoshenko, Vol. II, p. 423-426), the tension in combined bending and direct stress at B will be

$$S_{t\ max.} = \frac{Mh_1}{Aea} + \frac{P}{A} \qquad (1)$$

where M is the bending moment caused by P, A is the cross-sectional area, and dimensions h_1, e, and a are as shown in Fig. 1 (*b*). Likewise the compression at C will be

$$S_{c\ max.} = -\frac{Mh_2}{Aec} + \frac{P}{A} \qquad (2)$$

After making the substitutions

$$M = PR$$
$$h_1 = r - a$$
$$h_2 = c - r$$
$$e = R - r$$

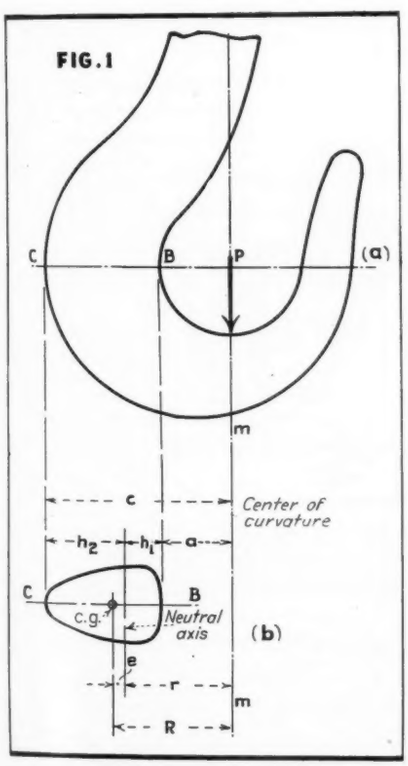


Fig. 1—Diagram for determining direct and bending stresses in hooks

and collecting terms, Equations (1) and (2) become

$$S_{t\ max.} = \frac{P}{A}\left[\frac{(R/a)^{-1}}{(R/r)^{-1}}\right] \qquad (3)$$

$$S_{c\ max.} = \frac{P}{A}\left[\frac{(R/c)^{-1}}{(R/r)^{-1}}\right] \qquad (4)$$

If it be desired to have the maximum unit stress in tension numerically equal to the stress in compression, the right hand sides of Equations (3) and (4) may be equated giving

$$\frac{2}{R} = \frac{1}{a} + \frac{1}{c} \qquad (5)$$

or

$$R = \frac{2\,ac}{a + c} \qquad (6)$$

This result gives the condition governing the location of the center of gravity which any type of cross section must satisfy in order that the maximum stresses may be numerically equal. Thus a cross-section in the form of a circle or parallelogram will not fulfill this condition. For such cross-sections

$$R = \left(\frac{a + c}{2}\right)$$

Consider the trapezoidal cross-section of Fig. 2. Dividing this into a parallelogram and triangle, the equation of statical moments about the center of curvature becomes

$$R\left(\frac{b_1 + b_2}{2}\right)(c - a) =$$
$$b_2(c - a)\left(a + \frac{c - a}{2}\right) +$$
$$(b_1 - b_2)\left(\frac{c - a}{2}\right)\left(a + \frac{c - a}{3}\right) \qquad (7)$$

Solving for R will give the location of the center of gravity.

$$R = \frac{b_1(2a + c) + b_2(a + 2c)}{3(b_1 + b_2)} \qquad (8)$$

If the maximum stresses are to be numerically equal, the right hand sides of Equations (6) and (8) are equated giving

$$b_1\left[2a + c - \frac{6ac}{a + c}\right] =$$
$$-\,b_2\left[a + 2c - \frac{6ac}{a + c}\right] \qquad (9)$$

or

$$\frac{c}{a} = \frac{2b_1 + b_2}{b_1 + 2b_2} \qquad (10)$$

Equation (10) gives the relationship which must hold between the various dimensions of the trapezoid. It is seen that there is a limiting position for the location of the center of curvature. Taking the extreme case, $b_2 = 0$, a triangular cross-section, the ratio $c/a = 2$. Thus the distance a must always be larger than the depth of the bar or $S_{t\ max.}$ will become numerically larger than $S_{c\ max.}$ for all trapezoids.

In a similar manner an equation of statical moments can be written for the T-shaped cross-section of Fig. 3. The various dimensions must satisfy the equation

$$\frac{4ac}{a + c} = \frac{b_1(d^2 - a^2) + b_2(c^2 - d^2)}{b_1(d - a) + b_2(c - d)} \qquad (11)$$

if the maximum and minimum stresses are to be numerically equal.

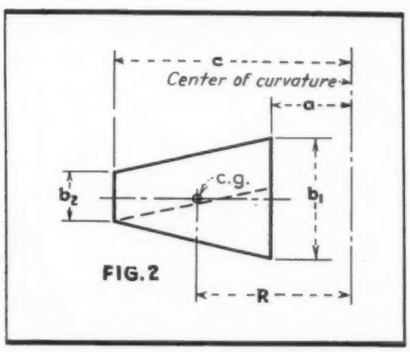


Fig. 2—Equation (10) gives relationships of dimensions for trapezoidal section

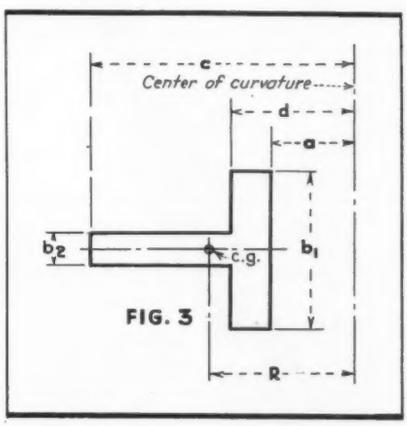


Fig. 3—Dimensions of T-shaped cross section must satisfy Equation (11)

Liquid Level Mechanisms —

Means of determining liquid level, detection of changes in liquid level, transmission of indicated levels, or warnings of changes beyond set limits; and means of using level changes for level control, or control of other conditions such as temperature and pressure, have been accomplished by numerous mechanisms. The most popular devices employ floats or pressure measurement with instruments such as the U-tube manometer, bourdon tube, and bellows.

The methods shown here are largely indicating methods or simple devices for automatic control of liquid level although they can conceivably be applied to control other conditions such as temperature and pressure. Methods using electric resistance of a column of liquid and measurement of pressure changes by means of piezo-electric crystals are not shown. Patent No. 2,162,180 describes a method involving determination of change in air pressure when a measured volume of air is introduced into a tank.

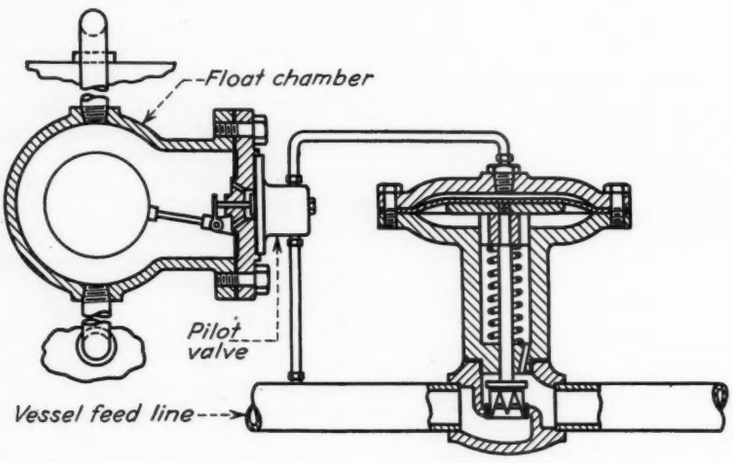


Fig. 1 - Float and Lever- Operated Pilot Valve

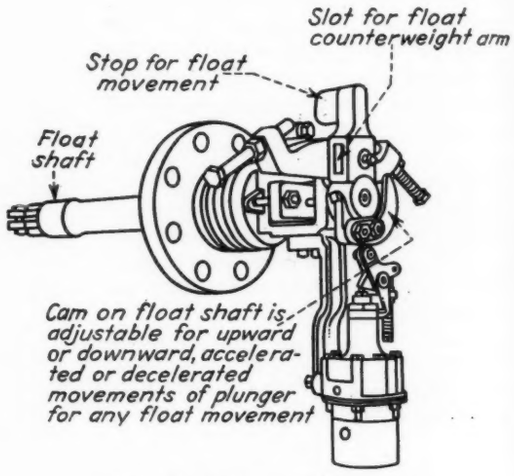

Slot for float counterweight arm
Stop for float movement
Float shaft
Cam on float shaft is adjustable for upward or downward, accelerated or decelerated movements of plunger for any float movement

Fig. 2 - Float and Cam-Operated Pilot Valve

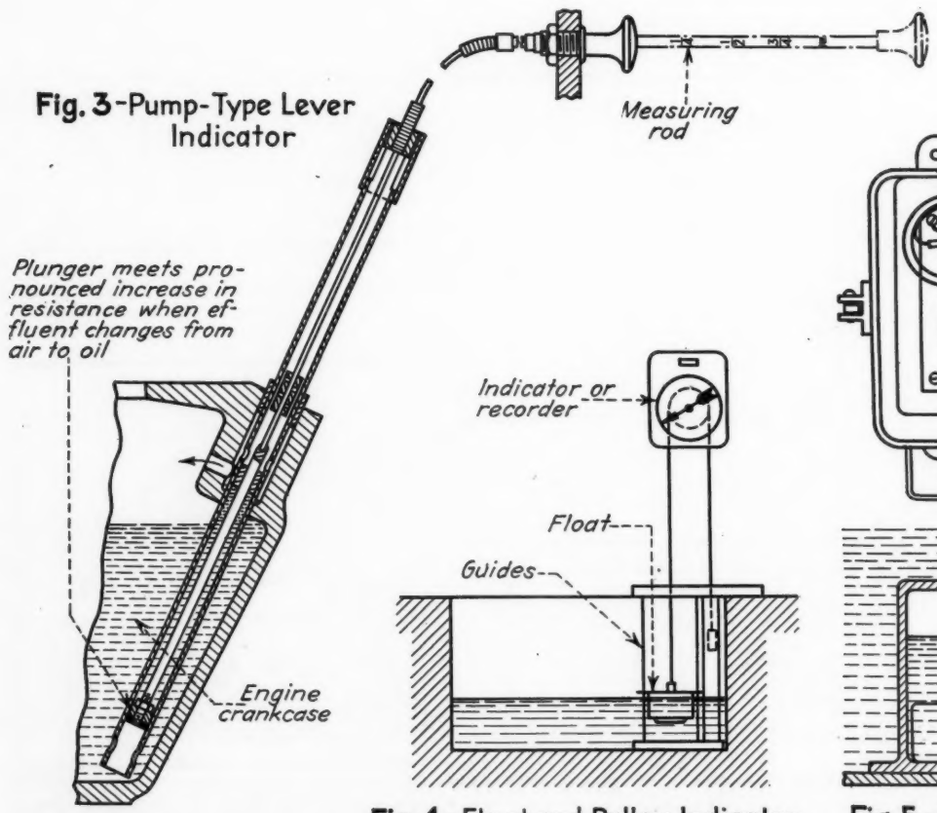

Measuring rod
Plunger meets pronounced increase in resistance when effluent changes from air to oil
Engine crankcase

Fig. 3 - Pump-Type Lever Indicator

Fig. 4 - Float and Pulley Indicator

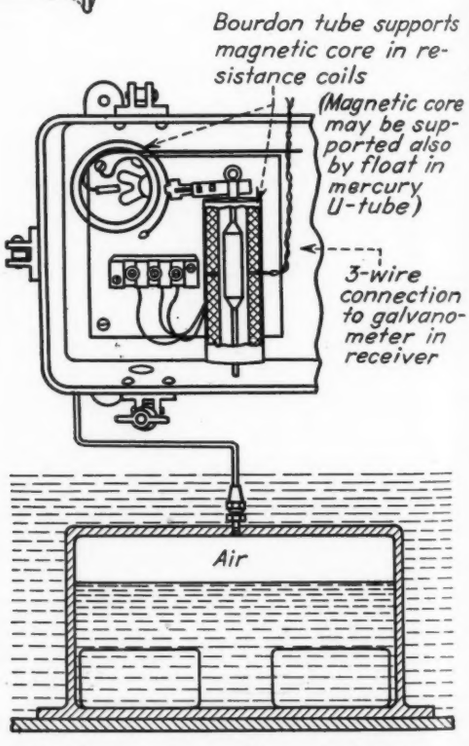

Bourdon tube supports magnetic core in resistance coils
(Magnetic core may be supported also by float in mercury U-tube)
3-wire connection to galvanometer in receiver
Air

Fig. 5 - Pressure Dome Indicator

Indicators and Controllers

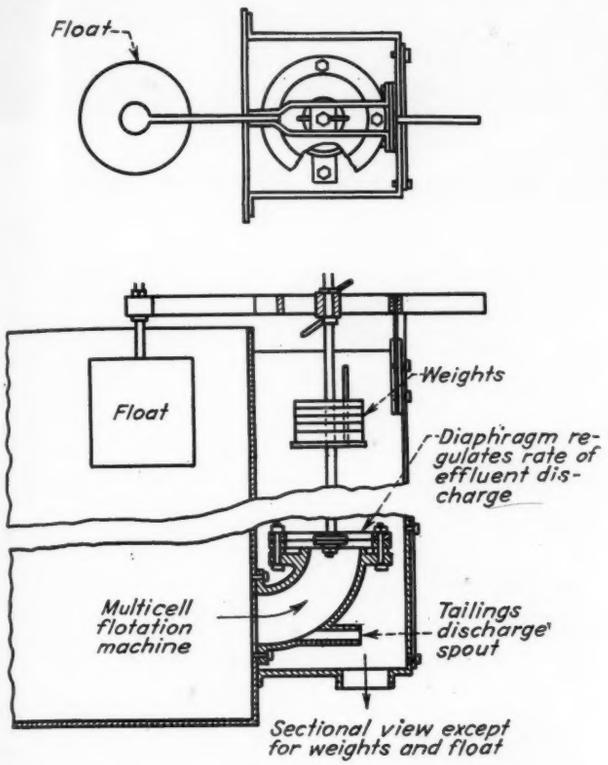


Fig. 6-Float Control of Discharge

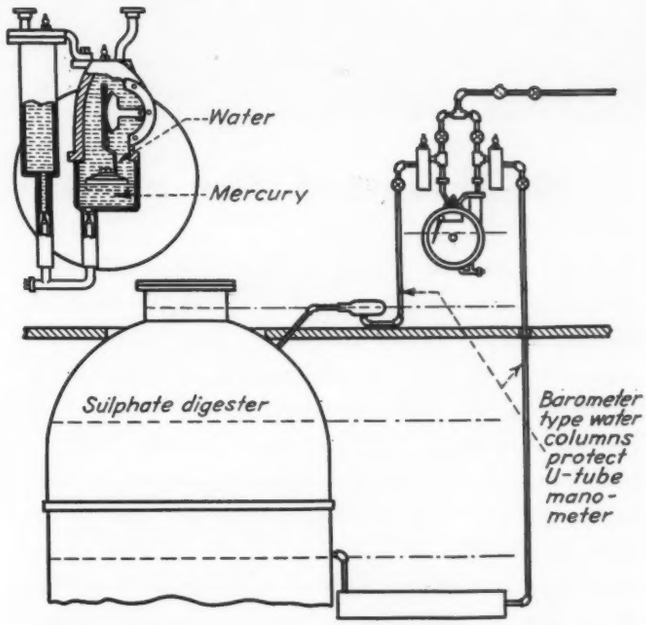


Fig. 7-U-Tube Manometer with Water Columns

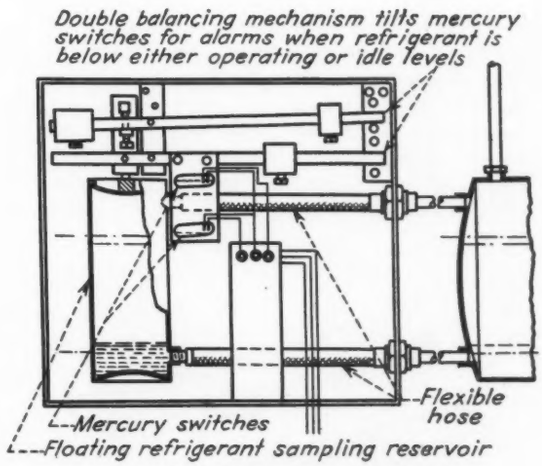

Double balancing mechanism tilts mercury switches for alarms when refrigerant is below either operating or idle levels
Flexible hose
Mercury switches
Floating refrigerant sampling reservoir

Fig. 10-Refrigerant Balance

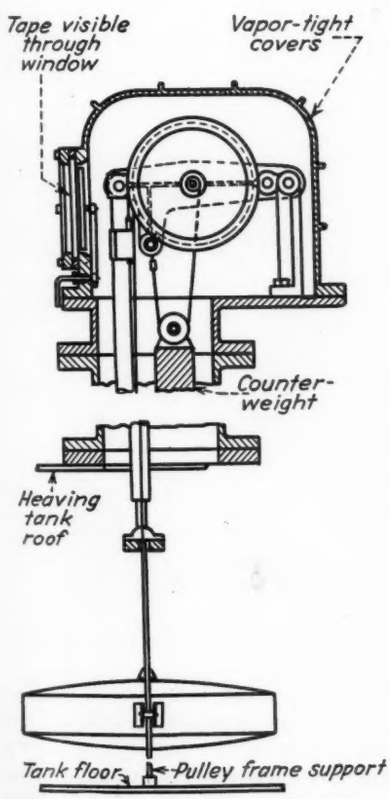


Fig. 8-Tank Roof Indicator

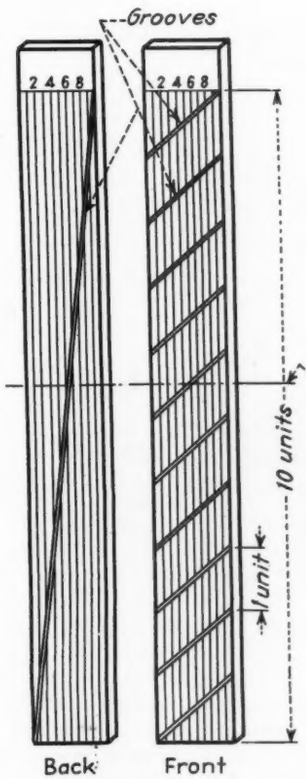


Fig. 9-Dip Stick Indicator

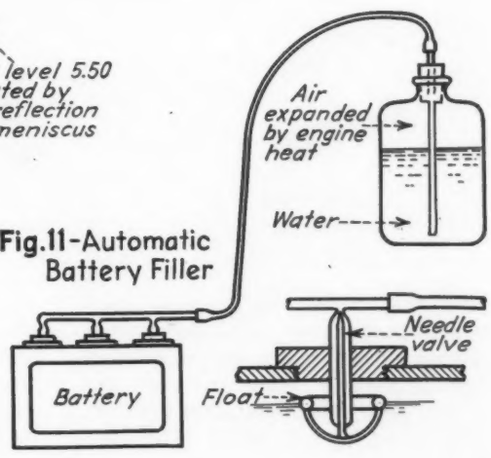


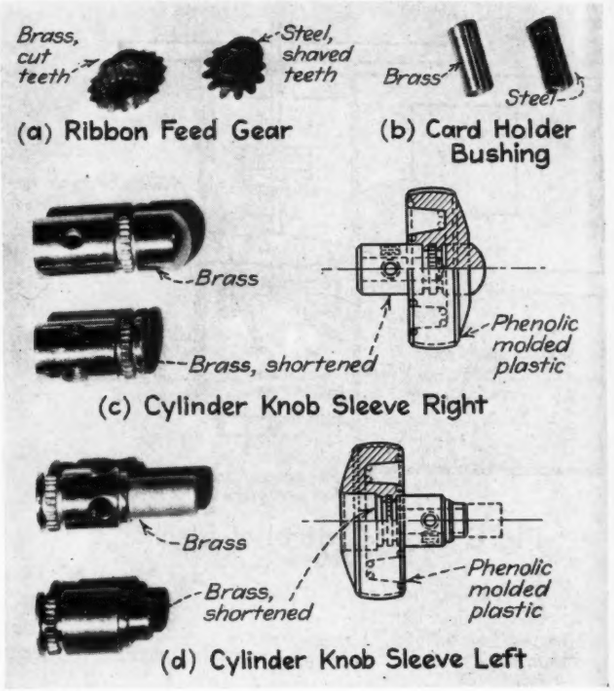
Fig. 11-Automatic Battery Filler

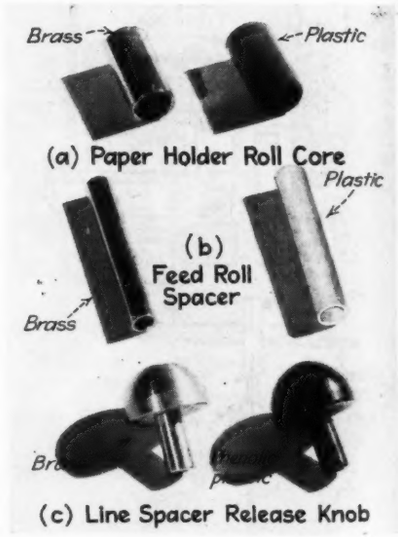
Modern Designs

Redesigned Royal Avoids Strategic Materials

Royal Typewriter Company in March 1941 initiated a design program with the sole purpose of eliminating strategic materials from their typewriters. Once a month design and production executives go over special inventory records and decide what steps must be taken to meet predictable materials shortages. So far Royal has "beaten to the draw" every acute shortage—in most cases making changes several months before necessary. Nickel consumption has been reduced from 3,000 lb. to 900 lb. per month. Brass is conserved by reducing size of parts or by switching to steel or plastics. All aluminum and zinc will be eliminated.

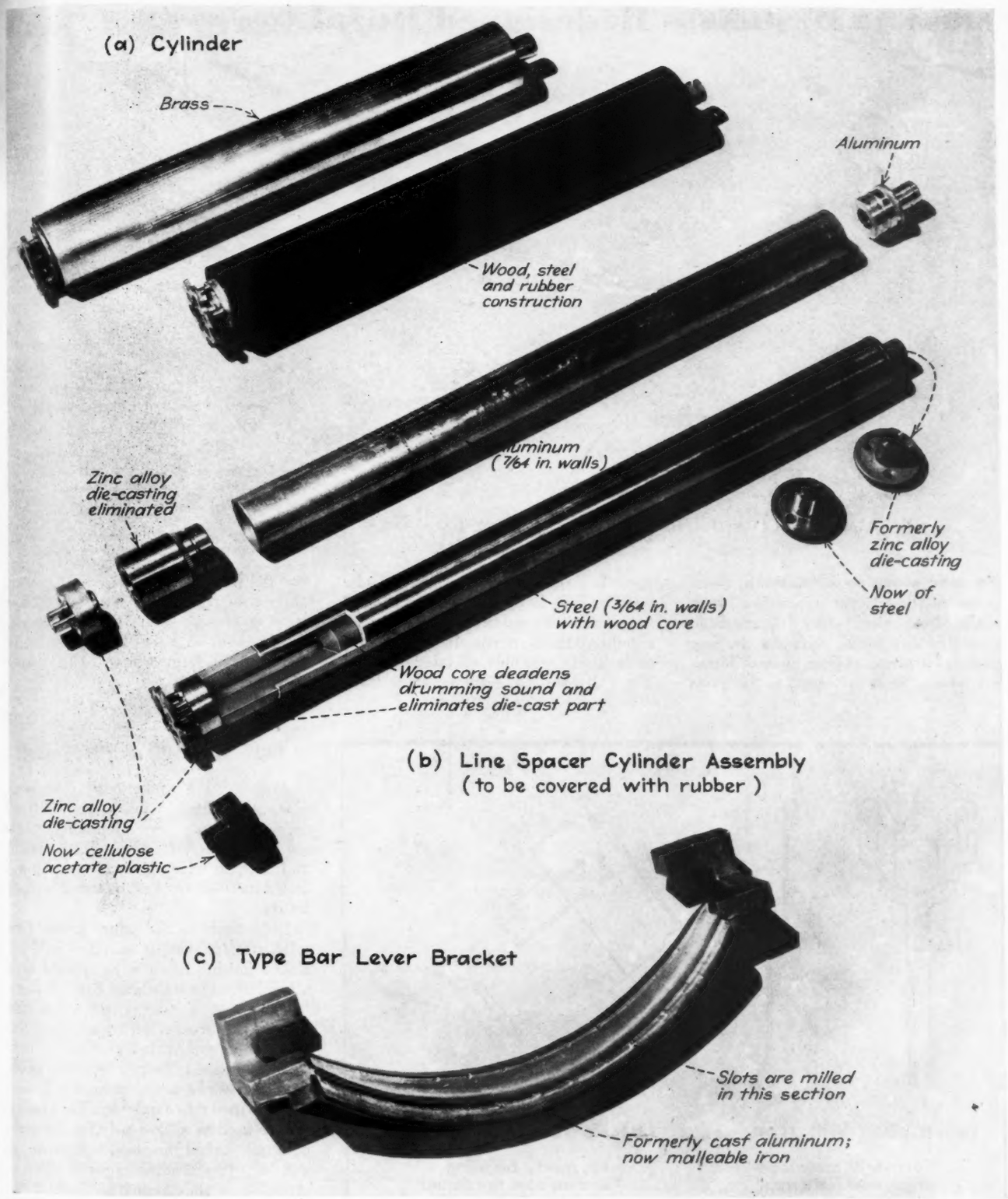
Brass to steel. Substitution of steel for brass in many high-production screw machine parts (such as shown in (a) and (b) at right) created a problem because screw-machine capacity, adequate for brass, was inadequate for steel at slower cutting speeds. Solution was to shorten some parts, as shown in (c) and (d), and to use plastics or steel for others. The part in (d) possibly may be eliminated entirely.





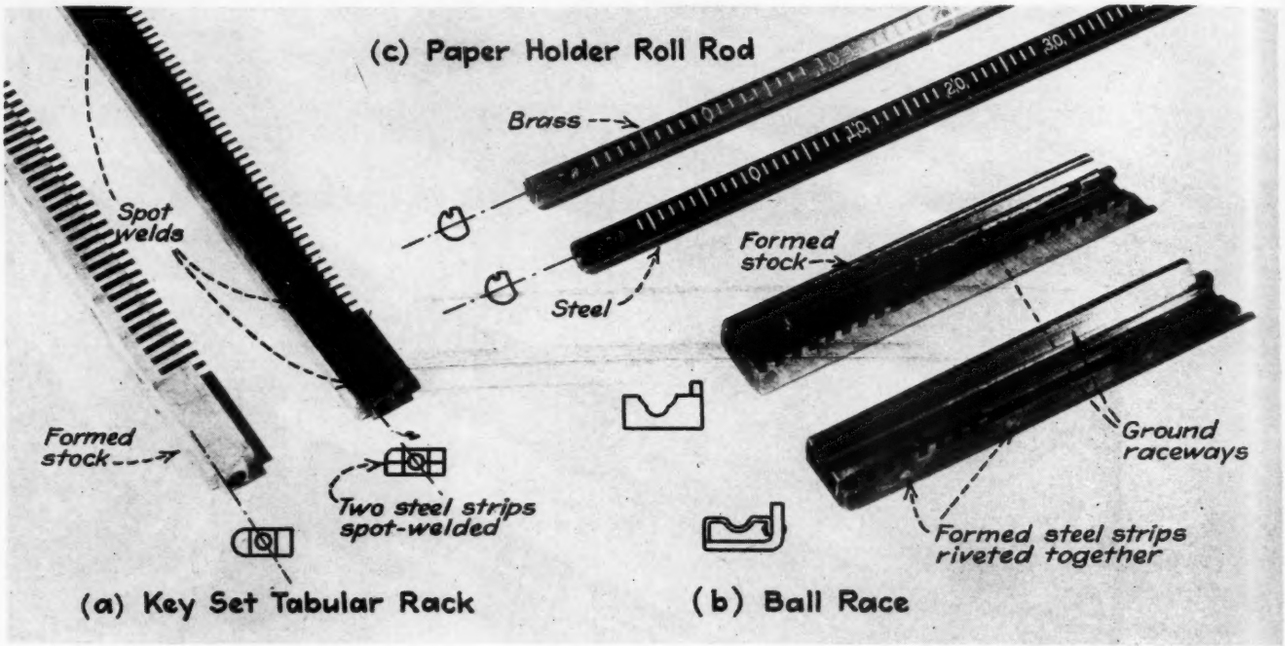
Porous bronze to plastic. Bushings formerly made of oil-impregnated porous bronze are now made of cellulose acetate plastics. Cellulose acetate was chosen after laboratory tests, both dry and oiled, equivalent to four years' service. Polystyrene shows excessive wear when dry; phenolic plastics swell up and bind when oiled. Aluminum also was eliminated in (a) above, and brass was eliminated in (b). Both parts were made of plastic at lower cost.

Brass to plastic. Parts shown above, now molded of plastic, were formerly cut from brass tubing or made on screw machines. Brass parts were chromium-plated, so shifting to plastics also saved considerable nickel.

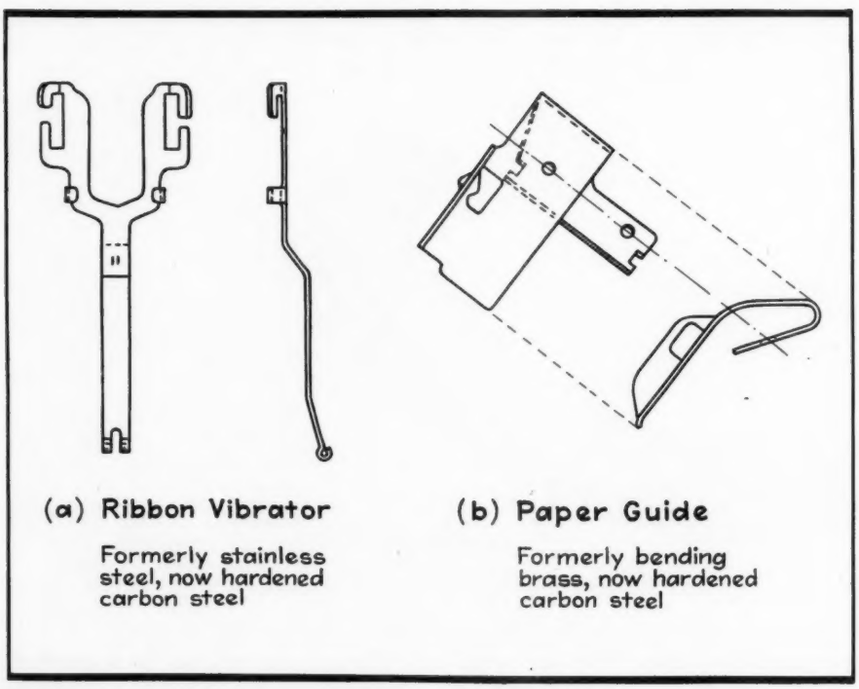

(a) Cylinder
Brass
Aluminum
Wood, steel and rubber construction
Zinc alloy die-casting eliminated
Formerly zinc alloy die-casting
Now of steel
Steel (3/64 in. walls) with wood core
Wood core deadens drumming sound and eliminates die-cast part
(b) Line Spacer Cylinder Assembly
(to be covered with rubber)
Zinc alloy die-casting
Now cellulose acetate plastic
(c) Type Bar Lever Bracket
Slots are milled in this section
Formerly cast aluminum; now malleable iron

Aluminum to steel and wood. Cylinder assemblies of Royal typewriters formerly consisted of parts made of aluminum, brass, and zinc alloy. These strategic materials have been entirely eliminated by redesigning the cylinder as a steel tube with a wood core. In early tests it was found that the thin-walled steel cylinder set up a drumming noise when the type keys struck the rubber covering. This was eliminated by extending the wood core all the way through. Smaller cylinders are made of wood without steel tube. Plastic cylinder end left and steel cylinder end right in present design are fastened to core with three wood screws. Approximately 25 cents was saved in the new design shown in (b).

Slots in the type bar lever bracket, shown in (c), are milled on special machines. The substitution of malleable cast iron for this part not only adds weight, but also necessitated the construction of new slot milling machines to replace the hand-operated aluminum slotters. (*continued on next page*)



Formed stock to steel strip. Some parts of the Royal typewriter were made from steel stock obtained in special drawn forms, reducing the machining required. These special forms are now difficult to obtain, so the parts are now made from standard strip, rods, or tubes. The key set tabular rack, (a), is now made by welding two steel strips together. One corner is slightly beveled to facilitate assembly of tabular stops. The ball race, (b), will be made from two formed steel strips riveted together. Holes are cut in the back of thicker strip to give access to the rivet heads. In (c) is shown a simple substitution of steel rod for formed brass in the paper holder roll rod.


(a) Ribbon Vibrator
Formerly stainless steel, now hardened carbon steel
(b) Paper Guide
Formerly bending brass, now hardened carbon steel

Stainless steel and brass to steel. Local annealing of one-quarter hard cold rolled steel permits bending the steel to sharp radii in the parts shown above. Parts are annealed over a special gas burner which directs flame jets at portions to be annealed. This substitution was made in several similar parts formerly of nickel steel, nickel silver, or brass hard stock. Nickel silver has also been replaced by straight chromium stainless steel.

Chromium Plating Reduced

Most of the nickel-base chromium plating on small visible parts of Royal typewriters has been eliminated. Substitutions are: attractive baked enamel and wrinkle finishes, tin electroplating, cadmium plating, and oxidized copper finishes.

Parts such as the paper guide (see left) are japanned or wrinkled finished. Baked finishes are always applied over a bonderized or tin-plated base. Bonderizing cannot be used on the graduated scales; the process seems to make the surface of the scales porous so that when the white is "wiped" into the graduations it sticks to the enamel and cannot be wiped off completely. Tin plating can be used as a base only for low-temperature baked finishes, otherwise it will "bead" and advantages of electroplating over tin dipping are lost.

Where a plated metal finish is desired, tin or cadmium are usually satisfactory. For some parts subject to scratches the hardness of cadmium is required, but for others the softer tin is adequate. These plated surfaces are not polished. Chromium plating is retained on such parts as the carriage return lever which are frequently handled and subject to abrasion and perspiration corrosion.

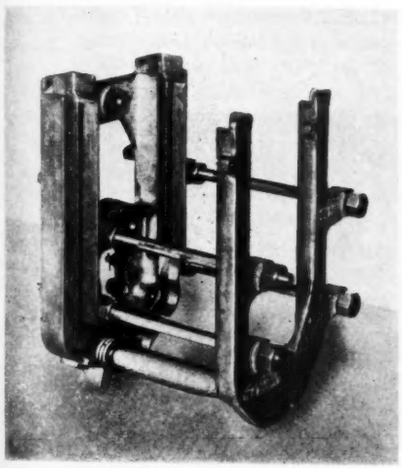
Cast Iron Replaces Aluminum in Loom Magazine

The magazine of a textile loom, the mechanism which stores the bobbins until they are fed into the shuttle, is connected to a selector motion which rocks the magazine so that the proper bobbin is fed to the shuttle at the right time. On Crompton & Knowles looms, high-speed operation of this magazine, which rocks about a bearing located near its center as shown in the illustration, makes light weight of the mechanism desirable. For this reason, the magazine had been designed to be made of aluminum castings, and had a total weight of about 10 pounds.

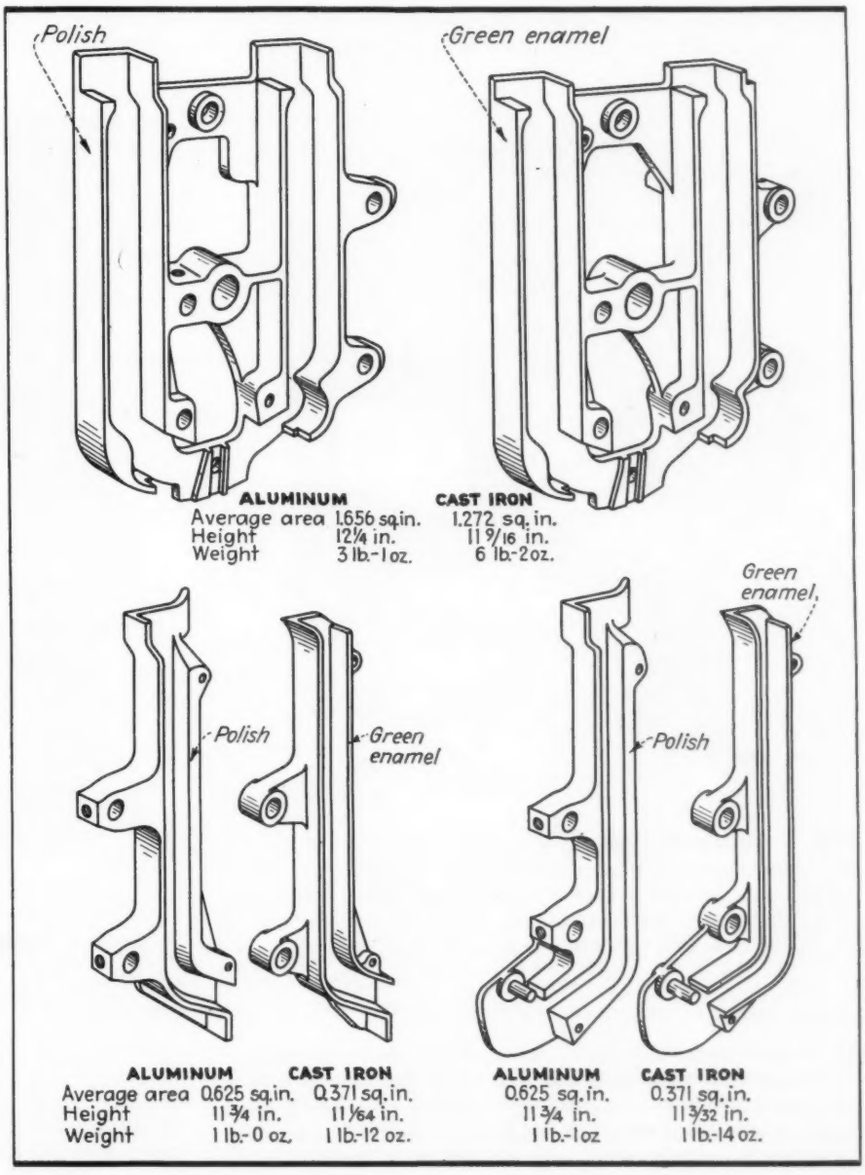
In addition to the requirement of light weight, the outside frame of the magazine must have a smooth, hard surface that will not corrode in an atmosphere of 75 percent humidity at temperatures averaging 82 to 85 deg. F. These surfaces come in contact with the thread filling on the bobbins. To meet this requirement in the aluminum frames, all outside surfaces were polished.

When aluminum grew scarce, Crompton & Knowles made a study to determine what substitute material could be used. Gray cast iron, for which the company maintained an excellent foundry with laboratory control, was the most logical material to consider. In order to use cast iron, it was necessary to reduce the weight of the castings to a minimum. This was accomplished by a study of the minimum cross sections possible with gray iron. This study showed that the assembled cast iron magazine would weigh only 1¾ times the aluminum magazine—considerably less than the expected 3:1 ratio. This acceptable increase in weight was compensated for by increasing the projected area of the supporting bearing by 36 percent.

The iron castings were finished with green enamel to meet the requirement for a hard, smooth, non-rusting surface. This finish is put on before machining. The changes in design, material, and finish are so satisfactory that aluminum may never again be used for C. & K. loom magazines. The cast iron parts are perfectly interchangeable, not only in the assembled mechanism, but also in the jigs and fixtures used to machine the three parts of the magazine frame.



Cast iron loom magazine, with one bobbin in place in one side, is illustrated above. The mechanism rocks on the bearing shown in the center of the left frame so that a bobbin from either side can be selected at the proper instant for feeding into the shuttle. The three iron castings of the frame are enameled to give a smooth, hard, corrosion-resistant surface.



Comparison of aluminum and gray iron for the three castings of the magazine is given in sketches at right. Minimum cross-sections with cast iron were calculated, from which approximate weights were determined. Cast iron mechanism, completely assembled, weighs only 75 percent more than the 10-lb. aluminum magazine. Note increased length of cast iron supporting bearing in the large casting to compensate for increased weight.

MODERN DESIGNS — Plain vs. Roller Bearings

Design of plain cast iron bearings has progressed in step with improved alloys, foundry practices, and lubrication. Today, manufacturers who have continued to use plain bearings find their experiences of great value in substituting plain bearings for roller bearings which are now difficult to obtain because of the National Defense Program.

Crompton & Knowles Loom Works have for years been producing looms designed so that they may be equipped with either type of bearing—and with equal success. Relative advantages of the two types, as set forth by Albert Palmer, director of engineering, and Stanley N. McCaslin, mechanical engineer are:

ROLLER BEARINGS:

1. Slight wear
2. Little attention to lubrication
3. Saving in power due to less friction of the rolling contact bearings
4. Interchangeability if replacement is necessary
5. Large weight supported for minimum amount of projected bearing area

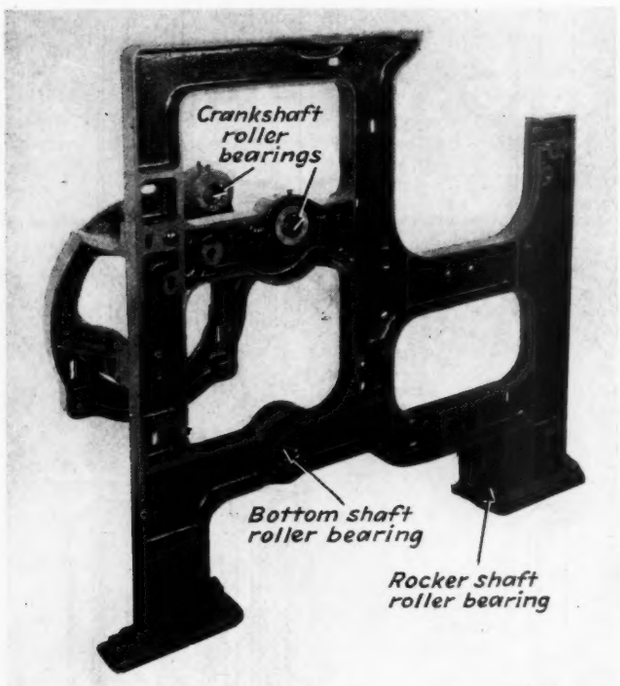
CAST IRON BEARINGS:

1. Low initial cost
2. Low replacement cost
3. With proper lubrication and material, long life due to large projected bearing area
4. Minimum outside diameter for given load
5. Uniform diameter for full length of shaft
6. Simpler to support

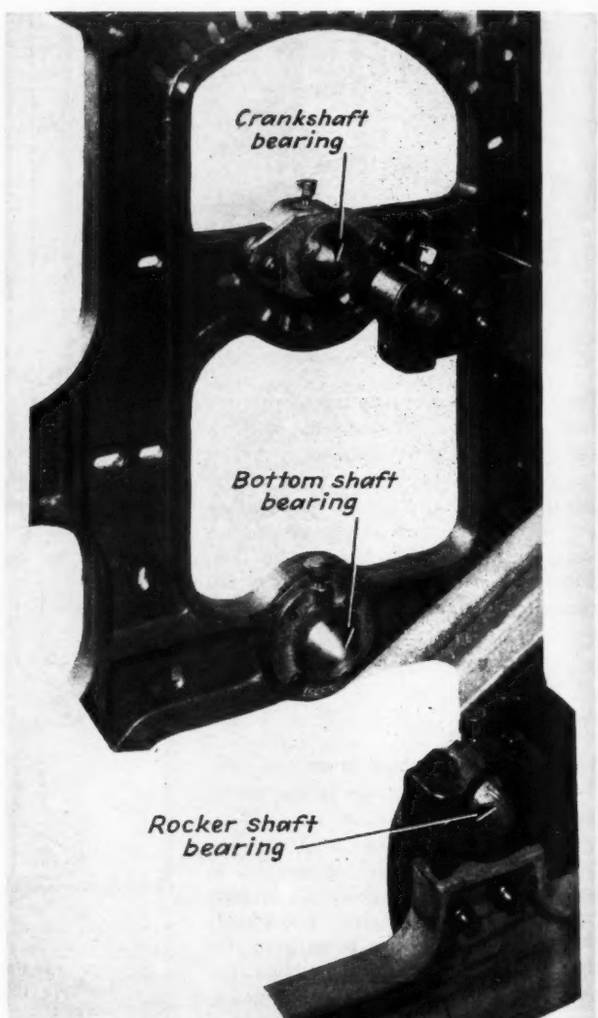
Main shafts of C. & K. looms run at maximum speeds of about 200 r.p.m., speeds at which cast iron bearings are quite satisfactory. The type of plain bearing used is the "cartridge" type, a construction which permits increasing the projected bearing area above that of a roller bearing installed in the same loomside casting.

Lubrication of plain bearings, a problem greater than for roller bearings, is by means of an oil cup and wick. Oil grooves in the bearings are straight. This method maintains a good lubricating film under the conditions of bearing load and speed, providing proper lubricant is used and oil cups are kept reasonably full.

To eliminate disadvantages of slight wear of plain bearings on some shafts such as the rocker shaft illustrated, a small roller bearing supplements the plain bearing. On shafts of smaller diameter, Crompton & Knowles sometimes specifies bronze bearings, which are easily replaceable. Oilless bearings, of solid wood or wood plug construction, are also specified for a few shafts.

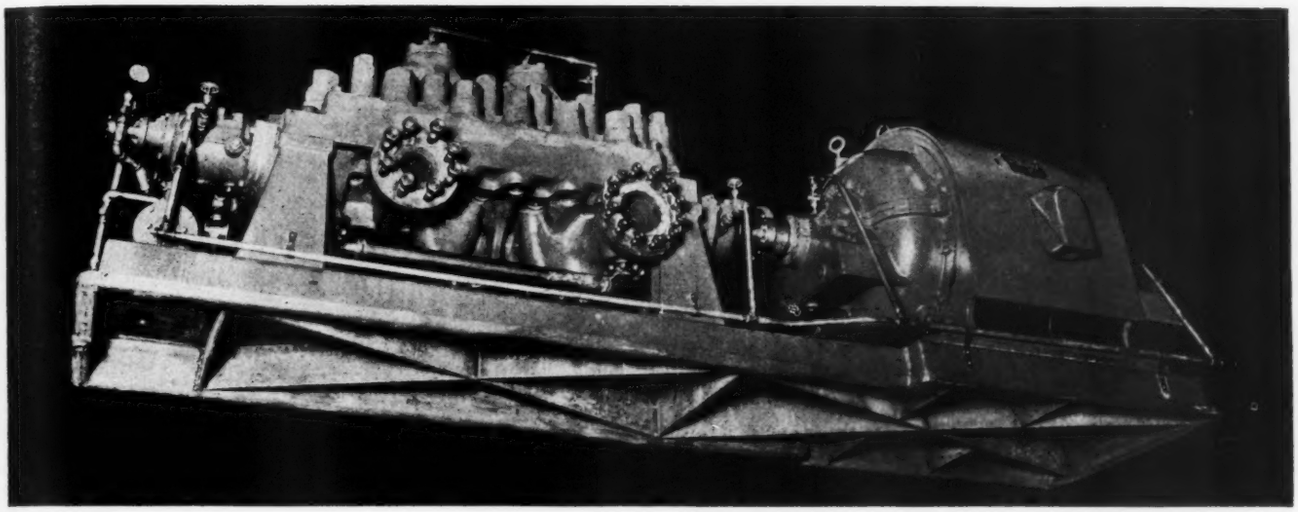


Roller bearings support three main shafts in the drive end loomside of an older type silk loom shown above. The crank shaft, located near the top, has two roller bearings, while the bottom shaft (at center) and rocker shaft (near the floor) have only one roller bearing at each end.

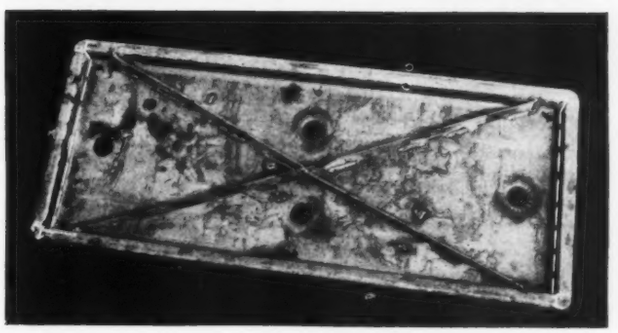


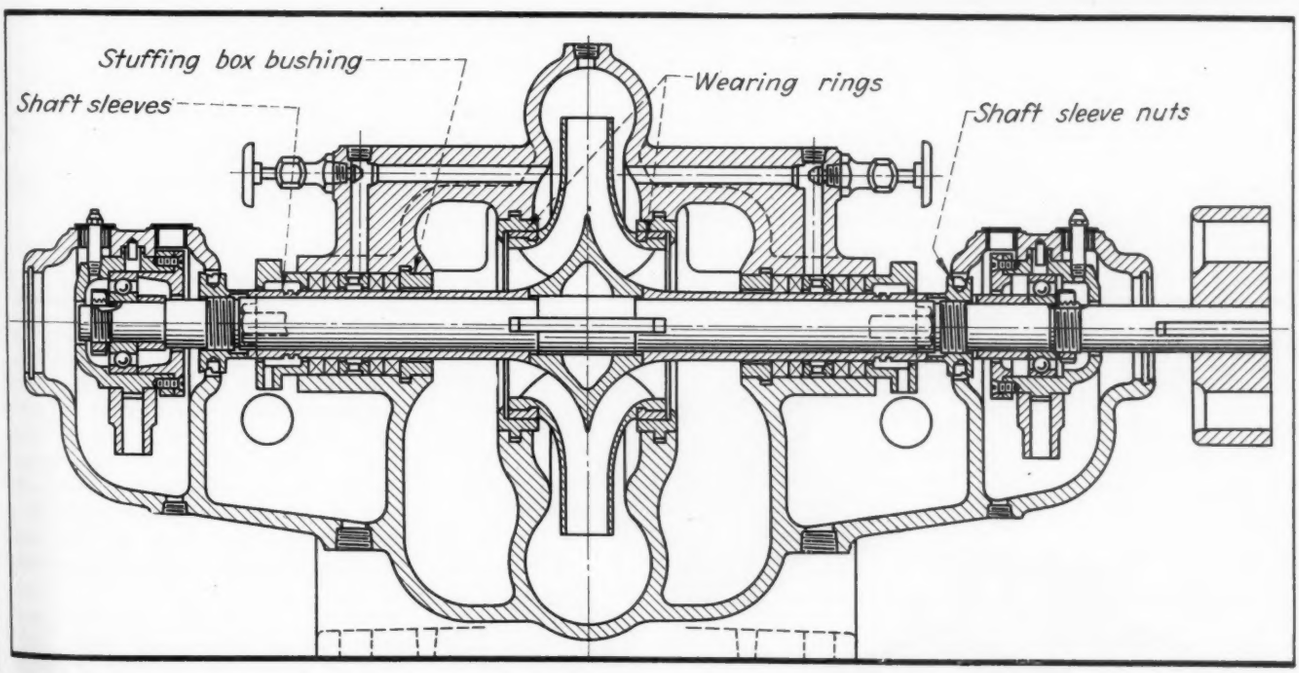
Plain bearings of the cast iron cartridge type are shown at right installed in the head end loomside of the new C. & K. silk loom. In this loom a small roller bearing is used on the rocker shaft (near floor) in addition to plain bearing to eliminate inaccuracy which may be caused by wear.

Centrifugal Castings and Welding



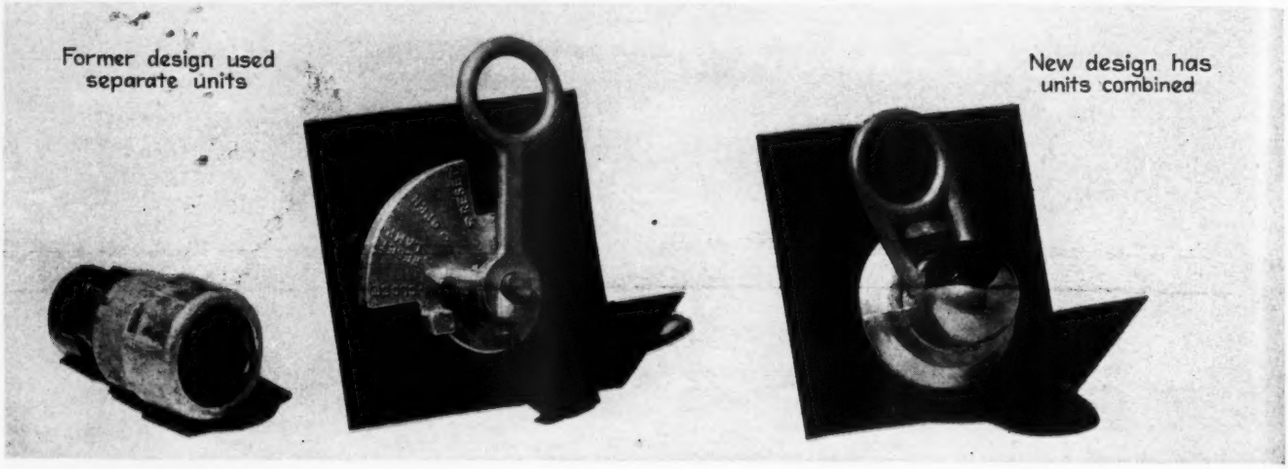
Welded steel base plates in Allis-Chalmers Type S centrifugal pumps have replaced castings with about 25 percent reduction in stock. With welding the pads for pump and motor can be placed to suit size of units thus making one base suitable for a variety of pump sizes. About $\frac{3}{8}$ in. plate is used making for light construction. Drip ledge is rolled and base made up in sections indicated by dotted line. Four corner pieces are welded in after ends have been welded to main plate. Cross members are welded inside the formed base to give added rigidity. At right is shown type of bracing and plate used with larger pump and motor, indicates possibilities of welded construction for unusual combinations.



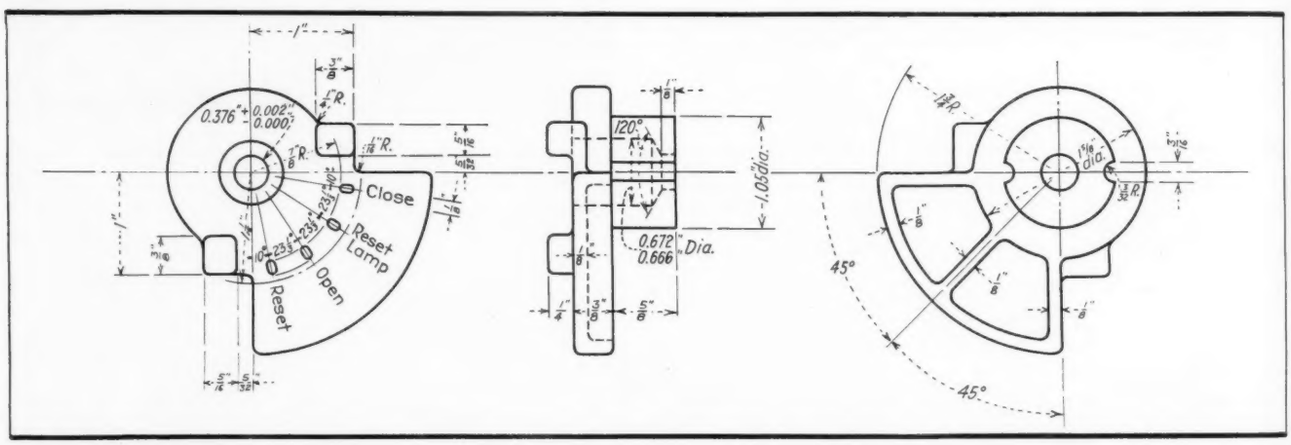


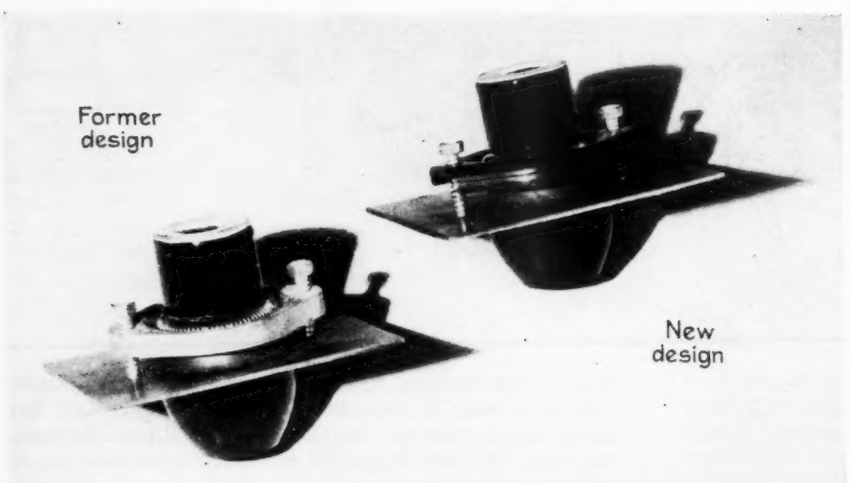
Machining time on these Type S centrifugal pump parts was reduced in direct proportion to the improvement in casting tolerances by changing from sand castings to centrifugal castings. Shaft sleeves, impeller wear rings, bushings and shaft sleeve nuts of aluminum bronze are now produced by a method developed by the company to $\frac{1}{16}$ in. tolerance or less. In addition to the reduction in machining costs it was found that the material was more uniform, Brinnel hardness was increased from 86 to 105 with improvement in tensile and elongation properties.

MODERN DESIGNS – Westinghouse Redesign



In the old design of mechanism used to operate the circuit breaker in distribution transformers, the mechanism consisted of a die-cast aluminum quadrant, detail design of which is shown above, together with various brass parts. This device is used in conjunction with a signal light, which formerly called for a number of aluminum die-cast parts and other pieces. The redesign of these two units combined them into one piece of equipment. The new design is better in appearance, reduces the space required on the tank walls, and assures the two elements being close together. Also, the new design has fewer parts, costs less to make, has better threads and, because they are made of brass, the material is somewhat more readily obtainable. The threads in the former aluminum die-castings were relatively poor as compared to the threads in the stronger and harder brass. Finally, appearance is further improved by having all of the parts on the outside wall of the tank made of the same material, namely, brass.





Flanges used for mounting porcelain bushings on the tank wall of distribution transformers were formerly of die-cast aluminum. There is appreciable stress in the flange when the flange screws are tightened against the tank wall. Aluminum has relatively low ultimate strength and elastic modulus and the die-castings were liable to be porous. The new design of flange is a steel stamping, considerably stronger and stiffer than aluminum and which, incidentally, avoids the stringent priorities restrictions now in effect on aluminum. The old design of bushing flange is shown on the left and the new pressed steel design on the right of accompanying illustration.

BESSEMER STEELS

Bessemer steel, now greatly improved in properties and uniformity, is in some ways superior to open hearth steel. It was more available than open hearth on July 15 when only 75 percent of bessemer capacity (8 percent total steel production) was operating. Tables by courtesy of C. C. Henning, Jones and Laughlin Steel Corp. Table IV based on curves by L. D. Woodworth, Carnegie-Illinois Steel Corp.

Table I—Qualities in Which Bessemer Steel Excels

	Machinability Percent	Stiffness Percent	Weldability Percent	Sensitivity Percent
Bessemer	100	100	100	100
Open hearth	75	85	90	80
Basis of rating	Commercial machining tests	Deflection tests	Hot welding, mainly lap and buttweld pipe	Marked change in physical properties by cold working

MACHINABILITY COMPARISON

	Rating
Bessemer screw (high sulphur)	120%
Bessemer screw (S.A.E. 1112)	100%
Open hearth manganese screw (S.A.E. X 1315)	85%
Open hearth screw (S.A.E. 1120)	70%
0.15 to 0.25 carbon open hearth low sulphur (S.A.E. 1020)	40%
0.10 to 0.20 carbon open hearth low sulphur (S.A.E. 1015)	40%

Greatest use for bessemer steel has been for screw machine steel, but it is now finding increasing use for structural purposes. Examples include railroad car sheathing, hot pressings, corner bands, door panels, and flat gussets. Its resistance to distortion makes it especially suitable for can bodies. Much wire is made of bessemer steel.

FATIGUE AND CREEP—The ratio of fatigue strength to elastic limit is approximately the same as for open hearth steel. Creep strength at normal temperatures is also about the same and allowable stress can be selected in the same manner as for open hearth. Lower fatigue life, however, disqualifies it for highly stressed parts.

WEAR AND CORROSION—Bessemer has good wear resistance for applications such as shaft journals. Copper-bearing bessemer steel resists atmospheric corrosion as well as open hearth copper-bearing steels.

METALLURGICAL QUALITY—Fine grain and fair resistance to grain growth characterize bessemer steel although grain size varies considerably. Bessemer steels are not heat-treated to the same extent as open hearth steels, except possibly screw steel, which is carburized and heat-treated in the manufacture of many small parts. In spite of its usual abnormality in grain size and high sulphur content, heat-treated bessemer makes an acceptable grade for use when extreme ductility of core is not essential.

STIFFNESS—In resistance to deflection, bessemer screw steel is stiffer than 0.40 to 0.50 percent carbon open hearth steels, the stiffness being further increased by cold working. Elastic modulus of bessemer is approximately 7 percent greater than that of open hearth steel of the same carbon content. Greater stiffness makes it harder to bend cold. It is likely to crack on cold bending and is also hot short at certain temperatures.

SHOCK LOADS—Bessemer steel is not generally used for load carrying members, especially those subjected to shock or impact. Its work hardening characteristics, relatively low fatigue strength, and greater stiffness tend to make it unsuited for such service.

ULTIMATE STRENGTH VS. CARBON CONTENT—Bessemer steel having the same carbon content as open hearth averages 15,000 lb. per sq. in. greater tensile strength. Conversely, for the same tensile strength bessemer steel requires 0.14 percent less carbon than open hearth steel. Thermal aging and increase in nitrogen content raise ultimate strength.

Table II—Bessemer Chemical Specifications

	Percentages			
	Carbon	Manganese	Phosphorus	Sulphur
Screw steel (standard grade)	0.08 to 0.16	0.60 to 0.90	0.09 to 0.03	0.100 to 0.180
Screw steel (high sulphur grade)	0.08 to 0.16	0.60 to 0.90	0.09 to 0.13	0.200 to 0.300
Screw steel (high carbon grade)	0.25 to 0.35	0.60 to 0.90	0.09 to 0.13	0.100 to 0.180
Skelp	0.08 max.	0.30 to 0.60	0.11 max.	0.08 max.
Bars, track spikes, sheet and tin plate	0.10 max.	0.30 to 0.50	0.11 max.	0.07 max.
Soft wire	0.12 max.	0.60 max.	0.11 max.	0.08 max.
Medium wire	0.10 to 0.20	0.70 max.	0.11 max.	0.08 max.
Reinforcing bars	0.15 to 0.35	0.70 max.	0.11 max.	0.08 max.
Structural soft	0.07 to 0.12	0.30 to 0.60	0.11 max.	0.08 max.
Structural medium	0.25 to 0.35	0.30 to 0.60	0.11 max.	0.08 max.
Structural high tensile	0.35 to 0.45	0.30 to 0.60	0.11 max.	0.08 max.

NOTE: Nitrogen is also believed to be significant in differentiating bessemer from open hearth steel.

WELDING QUALITY—Fusion and resistance welding qualities of bessemer steel are about the equal of open hearth. In certain tests involving over 300,000 tubes, failures of open hearth steel tubes were double those of bessemer tube failures. However, bessemer is extremely difficult to forge weld.

COLD FORMING—Bessemer steel cannot be cold formed. This type of steel cold works rapidly and will crack at the corners even when merely bending it to form angles. Bessemer should always be formed hot. Table V indicates that cold drawing the hot-rolled material increases the yield point of bessemer 75 percent and open hearth only 62 percent. Likewise it increases the tensile strength of bessemer 36 percent for bessemer and 20 percent for open hearth. Changes in elongation are nearly the same but cold working changed the reduction in area of bessemer 18 percent as compared with 8 percent for open hearth.

Table III—Physical Properties of Cold Worked Bessemer Screw Steel

Size In. Hex.	Cold Work Percent	Yield Point Lb. per sq. in.	Tensile Strength Lb. per sq. in.	Elongation Percent in 2 in.	Reduction Percent
½	None	45,000	65,740	28.2	60.6
15/32	12.4	86,170	91,200	18.0	56.0
7/16	23.6	96,120	102,400	12.0	43.0

Table IV—Comparison of Bessemer and Open Hearth Steels of Equal Ultimate Strength

Average phosphorus content: Bessemer—0.090%; Open Hearth—0.013%

Ultimate Strength Lb. per sq. in.	Steel	Yield Point Lb. per sq. in.	Elongation in 8 in. Percent	Izod Impact Ft. lb.
50,000	Bessemer		40.0	
	Open Hearth		34.7	
60.000	Bessemer	41,000	34.0	66
	Open Hearth	3 ,000	31.0	7
70,000	Bessemer	48,000	28.5	40
	Open Hearth	43,000	26.5	54
80,000	Bessemer	52,000	23.0	25
	Open Hearth	47,000	22.5	40

Table V—Effect of Cold Work on Bessemer and Open Hearth Steel

	Bessemer Steel		Open Hearth Steel	
	Hot Rolled	Cold Drawn	Hot Rolled	Cold Drawn
Yield point, lb. per sq in.	42,780	75,040	39,040	63,220
Tensile strength, lb. per sq. in.	64,900	88,170	65,000	77,880
Elongation, percent in 2 in.	38.0	19.5	36.5	19.0
Reduction, percent	65.2	53.3	61.8	57.0

While tensile strength is about the same after hot rolling, cold drawing increases the yield point of bessemer steel about 19 percent and the tensile strength 13 percent more than was accomplished on open hearth steel. When this sharp reaction to cold work is not desired it is possible to modify this characteristic by special deoxidation, as is done with open hearth steel.

PHYSICAL PROPERTIES OF BESSEMER STEELS—Test data given here are based on 1-in. diameter specimens in as-rolled condition without any form of heat treatment. Values should not be applied arbitrarily to a wide range of sizes because many variables such as size and processing during fabrication of finished materials affect test results. Elastic modulus of all grades ranges from 28,000,000 to 30,000,000. Tension test results are given as lb. per sq. in. cross sectional area. Breaking loads are not given.

BESSEMER SCREW STEEL

C. 0.08–0.16, Mn. 0.60–0.090, P. 0.09–0.13, S. 0.10–0.20
C. 0.08–0.16, Mn. 0.60–0.90, P. 0.09–0.13, S. 0.20–0.30

Approximate Physical Properties

	Hot Rolled	Cold Drawn
Tensile Strength, lb. per sq. in.	60,000 to 75,000	80,000 to 105,000
Yield Point, lb. per sq. in...	0.5 TS	70,000 to 90,000
Elongation (percent).....	25 min.	14 min.
Reduction (percent).....	45 min.	35 min.
Brinell.....	116 to 149	159 to 207
Izod (ft. lb. min.).....	30	.....

BESSEMER SCREW STEEL, HIGH CARBON

C. 0.25–0.35, Mn. 0.60–0.90, P. 0.09–0.13, S. 0.10–0.18
Bars in this grade are marketed on the basis of chemical limits alone.

BESSEMER SKELP

Physical tests rarely are made on bessemer skelp which is used for making furnace-welded pipe and welded couplings. In general, properties of skelp correspond to those for soft structural steel.

BESSEMER STEEL HOT-ROLLED MERCHANT BARS AND TRACK SPIKES

3. 0.10 max., Mn. 0.30–0.50, P. 0.11 max., S. 0.07 max.

Approximate Physical Properties

	Bars	Spikes
Tensile Strength, lb. per sq. in.	55,000 to 70,000	60,000 to 75,000
Yield Point. lb. per sq. in....	0.5 TS	0.5 TS
Elongation (percent).....	27 min.	25 min.
Reduction (percent).....	50 min.	45 min.
Brinell.....	109 to 137	116 to 149
Izod (ft. lb. min.).....	35	30

BESSEMER WIRE

Many variables make it impracticable to give general physical properties for this product. Wire, either bessemer or basic, usually is made under varying mill conditions to yield physical properties that are appropriate for the work to be performed by the consumer. The physical properties are affected appreciably by the size of the wire, the amount of cold working it receives and any heat treatment which may be required.

BESSEMER REINFORCING BARS

C. 0.15–0.35, Mn. 0.80 max., P. 0.11 max., S. 0.08 max

Approximate Physical Properties

	GRADE		
	Structural C.0.07–.15	Intermediate C.0.15–.30	Hard C.0.20–.35
Tensile Strength, lb. per sq. in...	55,000 to 75,000	70,000 to 90,000	80,000 min.
Yield Point, lb. per sq. in., ...	33,000	40,000	50,000 min.
Elongation (%)..	18 min.	14 min.	10 min.

BESSEMER STRUCTURAL STEEL, SOFT

C. 0.07–.12, Mn. 0.30–.60, P. 0.11 max., S. 0.08 max.

Approximate Physical Properties

Tensile Strength, lb. per sq. in.....	60,000 to 75,000
Yield Point, lb. per sq. in.....	0.5 TS
Elongation (percent).....	25 min.
Reduction (percent).....	50 min.
Brinell.....	116 to 149
Izod, ft. lb. min.....	30

BESSEMER STRUCTURAL STEEL, MEDIUM

C. 0.25–.35, Mn. 0.30–.60, P. 0.11 max., S. 0.08 max.

Approximate Physical Properties

Tensile Strength, lb. per sq. in.....	75,000 to 95,000
Yield Point, lb. per sq. in.....	0.5 TS
Elongation (percent).....	12 min.
Reduction (percent).....	40 min.
Brinell.....	149 to 192

BESSEMER STRUCTURAL STEEL, HIGH TENSILE

C. 0.35–.45, Mn. 0.30–.60, P. 0.11 max., S. 0.08 max.

Approximate Physical Properties

Tensile Strength, lb. per sq. in.....	80,000 to 110,000
Yield Point, lb. per sq. in.....	45,000 min.
Elongation (percent).....	10 min.
Reduction (percent).....	35 min.
Brinell.....	159 to 223

BESSEMER PROCESS IMPROVEMENTS—Modernization of bessemer steel quality has been in progress for several years and various investigational efforts have resulted in considerable improvement in the product. The improvement has been accomplished mainly by refinements in the control of the steel-making method and not solely by the adjustment in analysis. Practices better designed to minimize the amount of inclusions that are detrimental to machinability have resulted.

However, during the past several years the average machinability of bessemer screw steel grades has been improved upward of 15 percent and the uninformity of machining qualities from blow to blow has been greatly improved. Regulation of the predetermined ranges of physical properties has been improved more by a new automatic photocell method of control than by any advance in bessemer metallurgy in its history, according to one authority still further improvements in machinability are expected.

A number of attempts have been made to improve bessemer steel through adjustment in analysis, particularly directed toward the reduction of phosphorus. However, there is some uncertainty regarding the importance of the presence of this and other elements, as evidenced by the fact that open hearth steel rephosphorized to the same analysis as bessemer does not have bessemer characteristics. Oxidation in the bessemer process may be the source of quality difference and nitrogen may be the most significant element. Investigation in this direction has not been as satisfactory as those directed toward closer control of the process.

ALTERNATE STEELS

CONFRONTING THE ENGINEER in the problem of choosing one steel to take the place of another is the knowledge that minor changes in chemical combinations may cause major changes in one or more physical properties. Steels with identical tensile properties do not necessarily have the same impact, distortion or hardenability characteristics. The differences in specific properties of the steel customarily used and the alternate steel selected will determine the extent of compromise necessary in the design and the method of producing the part. For many applications an alternate steel can be found that is as satisfactory as the steel which has been replaced.

For some parts of large section in which mass properties are a consideration, the effect of nickel may be difficult to duplicate satisfactorily, so that when alternates are used the part will have to be radically redesigned to suit the properties of the steel that has been selected.

Properties which are imparted to steel by heat-treatment effects produced in operations such as torch-cutting, or hot forming and drawing, should be studied thoroughly to determine what changes in heat-treating methods would be necessary to adapt the alternate material for production operations. Other factors that should be analyzed carefully are decarburization, distortion, fatigue properties, impact characteristics and temperature requirements in service.

The tables and charts presented on these pages deal only with constructional steels that contain not more than 1.50 percent of chromium, 5.25 percent of nickel, 0.75 percent of molybdenum, 2.00 percent of manganese, 2.00 percent of silicon, and 0.20 percent of vanadium. The data have been compiled from the publication, "Possible Substitutes for Nickel Steels," issued by the American Iron and Steels Institute. The values given for the various steels listed in the charts are representative of typical physical properties obtained from samples of one inch diameter except where otherwise noted. These values should not be used to set specification limits, since new data are constantly being assembled and as new experience is gained the published data may have to be modified.

Possible Alternates for Nickel Steels

In Current Use	Possible Alternate
Nickel steel series designation	Standard steel series designation
2300	4600, 3100, 4000, 4100, 6100
2500	3300, 4800, 4300, *4300
3000	4000, 4100, 5100, 6100
3100	4000, 4100, 5100, 6100
3200	3100, 4000, 4100, *4100, 6100
3300	*4300
3400	*4300
4300	No alternate suggested
4600	3100, 4000, 4100, *4100, 6100
4800	4300, *4300

* These steels contain a higher molybdenum content than the regular 4300 and 4100 series. They are still in the development stage, but research work is being done to study further their possible application.

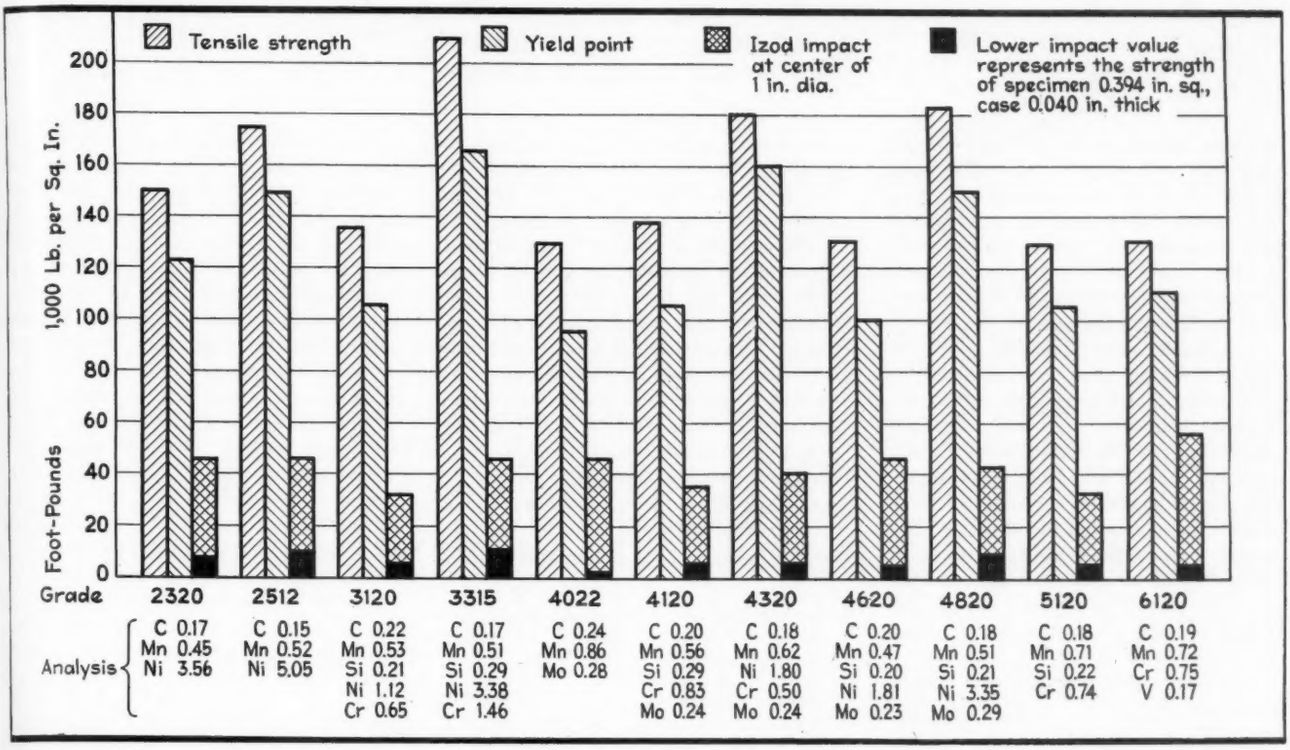
Comparative Physical Properties of 0.40/0.50 C. Grades

Oil quenched from the optimum temperature for each grade
All impact values given for room temperature

Grade Designation	Specimens 1 in. dia. Drawn to 300 B.H.N. approx.				Specimens 1 in. dia. Drawn to 400 B.H.N. approx.				Tension specimens 0.505 in. dia. Impact specimens 0.394 in. dia. Fully quenched, drawn at 400 deg. F. to 500/550 B.H.N.			
	Tensile Strength 1,000 lb. per sq. in.	Yield Point 1,000 lb. per sq. in.	Red. Area per-cent	Izod Impact ft. lb.	Tensile Strength 1,000 lb. per sq. in.	Yield Point 1,000 lb. per sq. in.	Red. Area per-cent	Izod Impact ft. lb.	Tensile Strength 1,000 lb. per sq. in.	Yield Point 1,000 lb. per sq. in.	Red. Area per-cent	Izod Impact ft. lb.
2345	140	130	58	40	194	180	48	18	267	230	32	15
X-3140	144	120	57	50	206	180	46	18				
3145	150	130	55	46	208	180	44	12	260	220	24	8
3240	144	122	57	57	200	176	47	22	275	240	37	11
3340	150	130	57	50	218	188	47	15				
4065					194	175	38		274	238	24	
4140	155	128	59	55	215	185	45	12				
4145									275	235	24	7
4340	148	130	60	72	206	185	52	14				
4640	146	125	56	45	195	177	50	22				
4645									270	230	38	15
5150	150	136	58	55	200	175	50	17				
6140	144	135	54	75	194	185	52	18				
6145									275	235	23	8

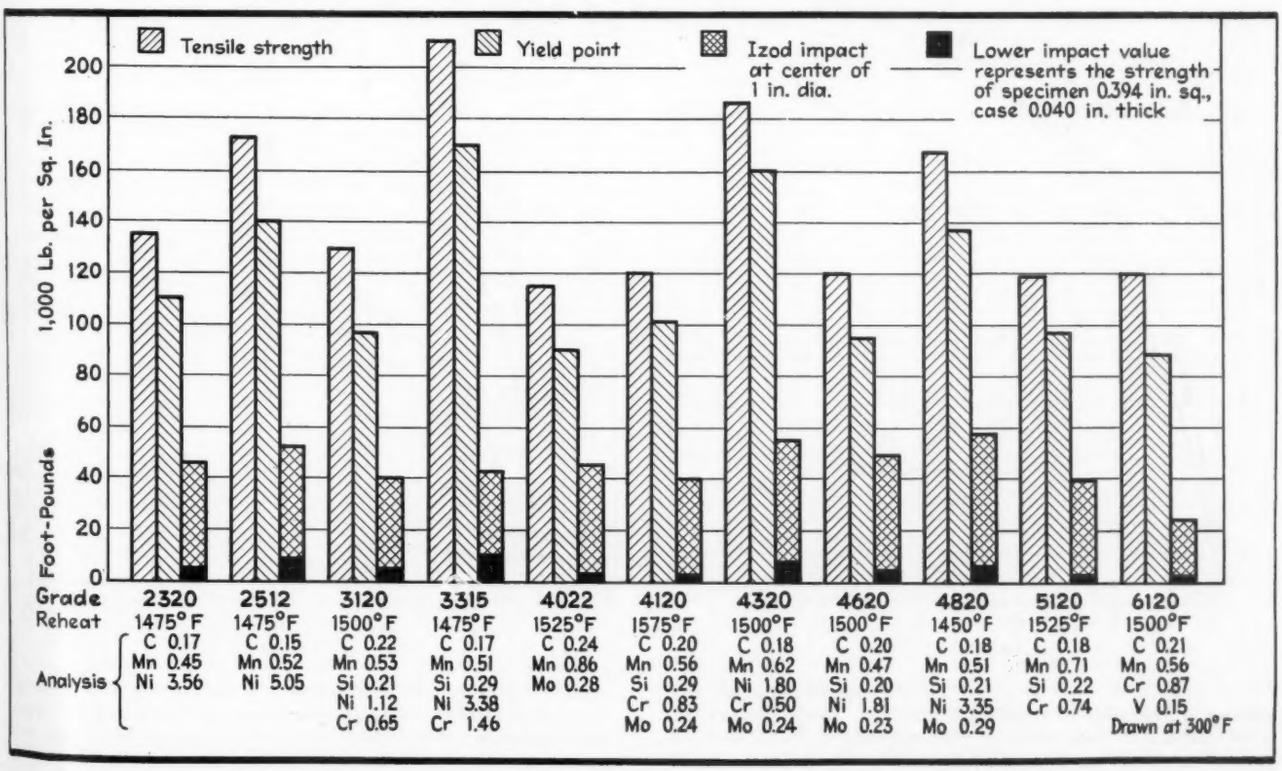
Carburizing Steels — Direct Quenched

Center specimens 1 in. dia. pseudo-carburized 8 hr. at 1,680 deg. F. Oil quenched, Drawn 1 hr. at 400 deg. F.



Carburizing Steels — Reheated to Obtain Optimum Physical Properties

Center specimens 1 in. dia. pseudo-carburized 8 hr. at 1,680 deg. F., oil quenched. Reheated as indicated, oil quenched, drawn 1 hr. at 400 deg. F. Temperatures given are optimum for max. Izod value carburized samples



GILDING METAL PROPERTIES

R. A. WILKINS and E. S. BUNN
Revere Copper and Brass Incorporated

GILDING METAL is an alloy of copper and zinc containing 95 percent copper. It should not be confused with the gilding metal mentioned in government specifications and known commercially to the brass trade as "commercial bronze," which contains 90 percent copper, and 10 percent zinc.

The charts show mechanical properties of gilding metal as influenced by cold working and annealing treatments. In both processes the properties are given for two "ready-to-finish" grain sizes, 0.015 and 0.070 mm., which represent limits commonly encountered in commercial annealing of this alloy. The anneal before final cold working is called "ready-to-finish anneal" and the grain or crystal size produced in this anneal is called "ready-to-finish grain size."

The mechanical properties of gilding metal are markedly influenced by the ready-to-finish grain size. This is especially true for the cold worked series shown by Figs. 1-4. When the cold work is followed by a final anneal, as shown in Figs. 5-9, the influence of the ready-to-finish grain size is less marked but nevertheless as important.

The amount of cold work performed is commonly indicated by B. & S. numbers hard. All brass sheet and strip thicknesses are based on Brown and Sharpe Gauge units. If a piece of brass is cold rolled from 12 B. & S. gauge to 13 B. & S. gauge it is said to be 1-number hard, of from 12 to 16 B. & S. gauge —4-numbers hard. The relation between numbers hard, percentage reduction of area by cold working and commercial temper designations is given in the following table.

Commercial Temper Cold Rolled Sheet and Strip	Numbers Hard	Percent Reduction of Area
Quarter Hard	1	10.9
Half Hard	2	20.7
Three Quarter Hard	3	29.4
Hard	4	37.1
Extra Hard	6	50.0
Spring	8	60.5
Extra Spring	10	68.7

A different range of properties is of course established for each number of hardness, as shown in Fig. 1-4. This range is typical of commercial production and as such is recognized in commercial specifications. Control of the ready-to-finish grain size is, therefore, essential. Fig. 1 shows that material having a ready-to-finish grain size of 0.015 mm. and cold worked 3-numbers has the same tensile strength as material with a ready-to-finish grain size of 0.070 mm. cold worked 5-numbers hard.

The influence of ready-to-finish grain size on the properties of the material at various temperatures of final anneal following cold work to 6-numbers hard (50% reduction), is shown in Figs. 5-9. Material having a ready-to-finish grain size of 0.070 mm. has consistently lower mechanical properties for all final annealing temperatures than material having a ready-to-finish grain size of 0.015 mm. Grain growth takes place more rapidly in the material having the larger ready-to-finish grain size.

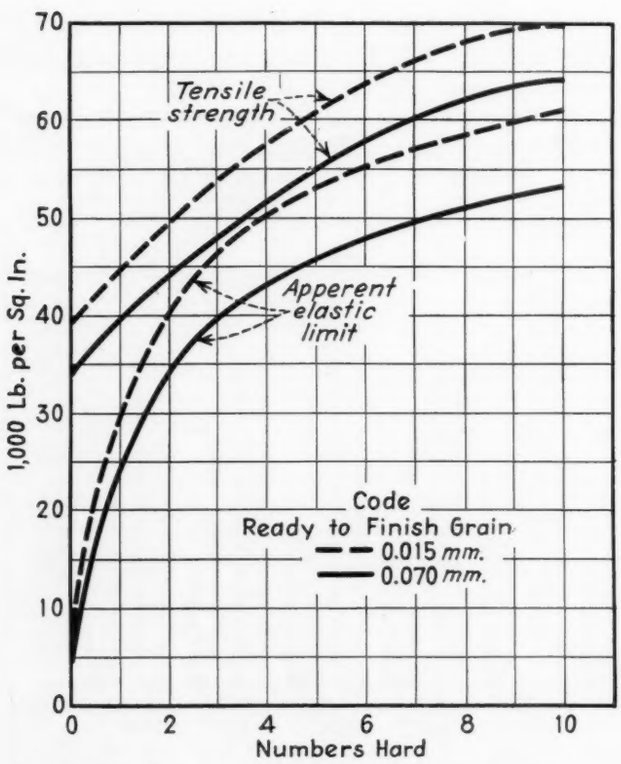


Fig. 1—Effect of cold working on the tensile strength and apparent elastic limit of gilding metal. The space between the solid and dash lines indicates the influence of ready-to-finish grain size on the physical properties

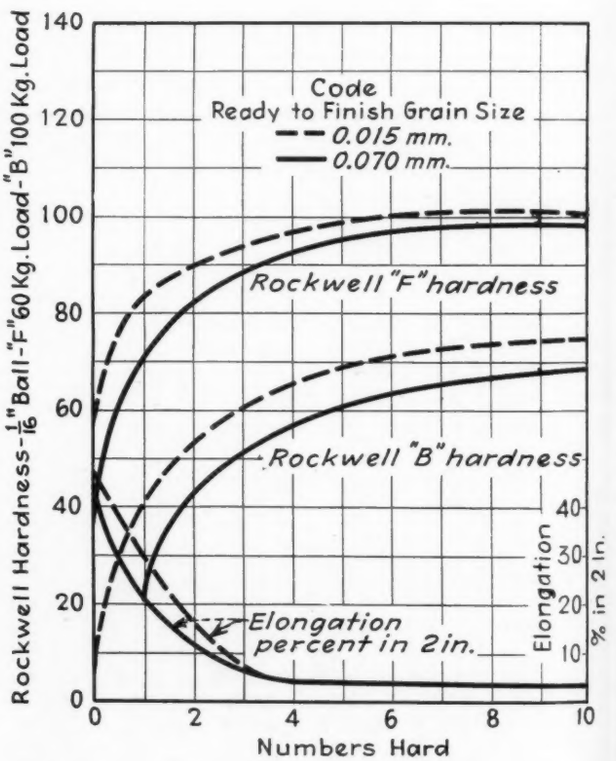


Fig. 2—Effect of cold working on the hardness and percentage elongation in two inches of gilding metal. Ready-to-finish grain size determines the position of a curve between the solid and dash lines for limiting ready-to-finish grain sizes

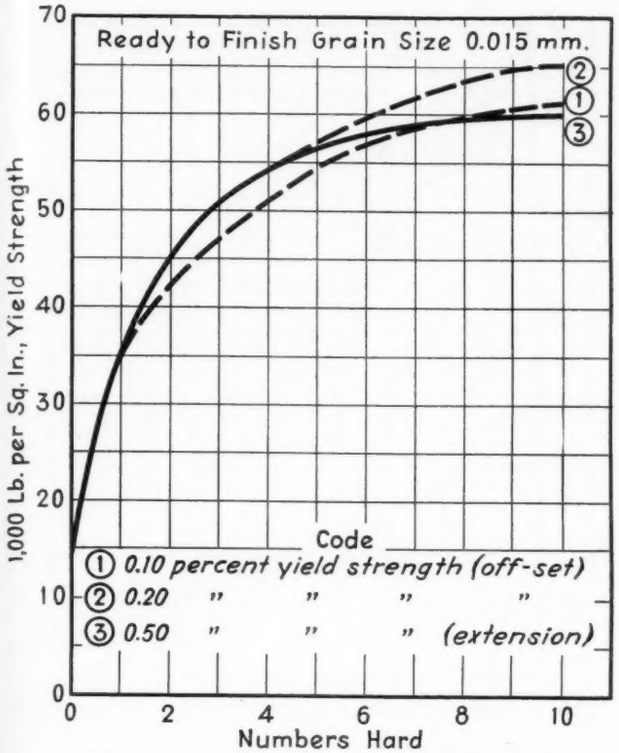

Ready to Finish Grain Size 0.015 mm.
1,000 Lb. per Sq. In., Yield Strength
Numbers Hard
Code
① 0.10 percent yield strength (off-set)
② 0.20 " " " "
③ 0.50 " " " (extension)

Fig. 3—Effect of cold work on the yield strength of gilding metal having a ready-to-finish grain size of 0.015 mm. The influence of ready-to-finish grain size on yield strength for a given amount of cold work can be compared in Figs. 3 and 4

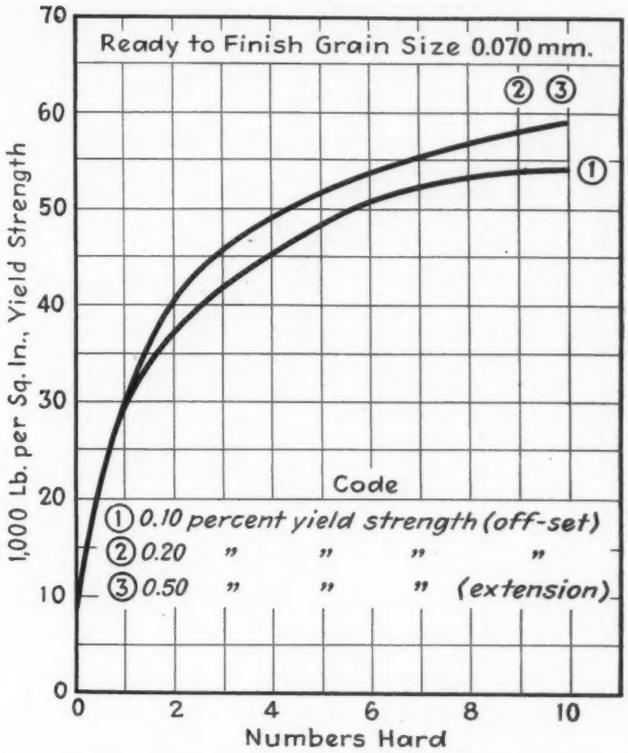

Ready to Finish Grain Size 0.070 mm.
1,000 Lb. per Sq. In., Yield Strength
Numbers Hard
Code
① 0.10 percent yield strength (off-set)
② 0.20 " " " "
③ 0.50 " " " (extension)

Fig. 4—Effect of cold work on the yield strength of gilding metal having a ready-to-finish grain size of 0.070 mm. The influence of ready-to-finish grain size on yield strength for a given amount of cold work can be compared in Figs. 3 and 4

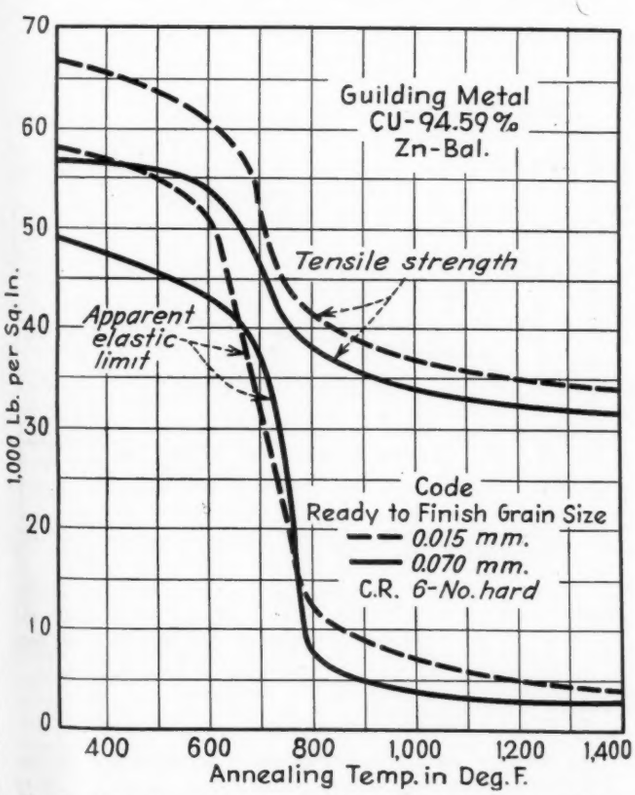


Fig. 5—Effect of a final anneal after cold rolling to 6-numbers hard (50 percent reduction). Space between the solid and dash curves still shows the influence of ready-to-finish grain size. The greatest change in physical properties takes place between 600 and 800 deg. F.

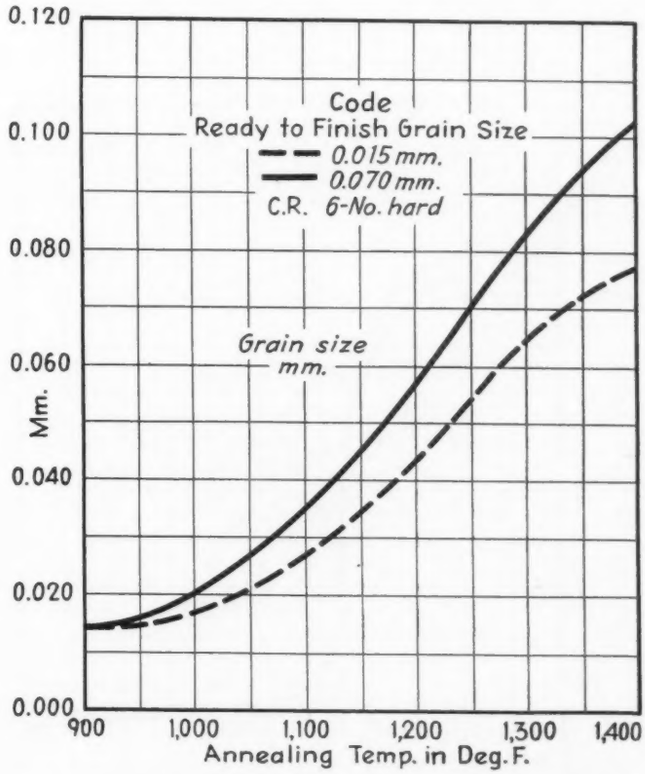


Fig. 6—Effect of ready-to-finish grain size on grain growth during the final anneal, which followed cold rolling to 6-numbers hard, is evident by the spread of the two curves. No recrystallization takes place below 700 deg. F. Up to 900 deg. F. grain growth is too slow for satisfactory measurement

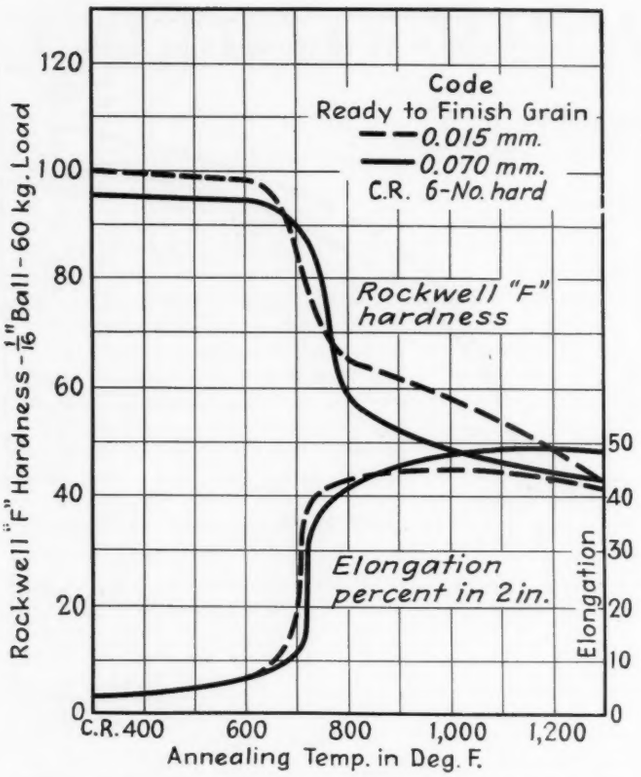


Fig. 7—Effect of change in annealing temperature (one hour at temperature) on hardness and percentage elongation on materials of two ready-to-finish grain sizes at the common limits of size, after cold work to 6-numbers hard

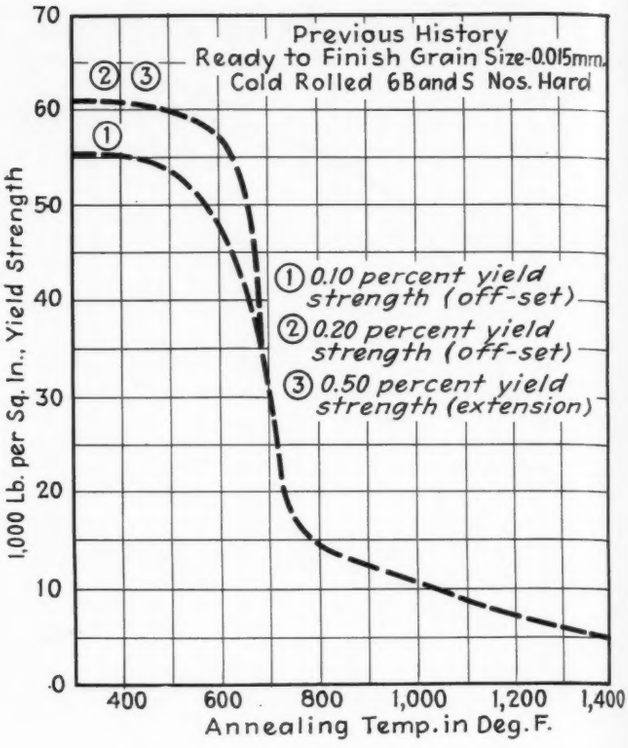


Fig. 8—Effect of annealing on yield strength of gilding metals of limiting grain size after cold work to 6-numbers hard. The influence of ready-to-finish grain size for a given annealing temperature can be compared in Figs. 8 and 9

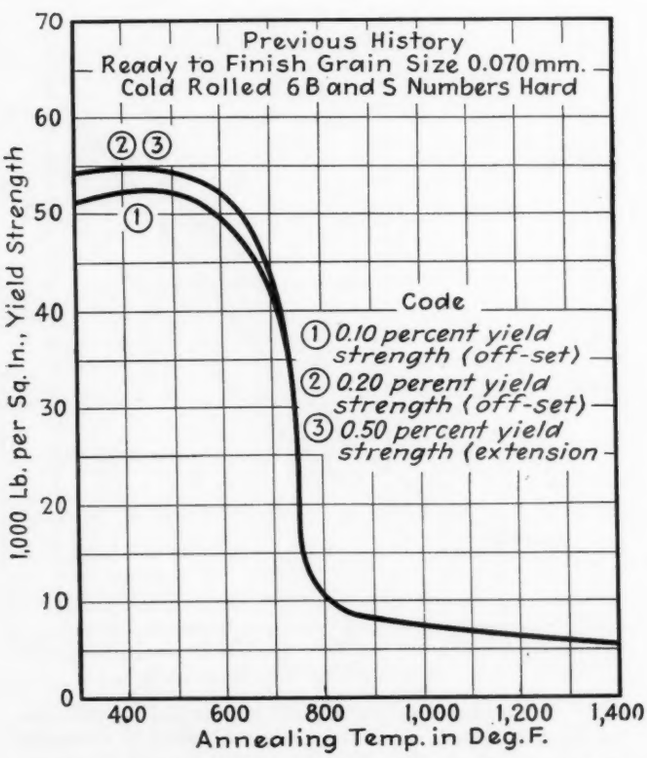


Fig. 9—Effect of annealing on the yield strength of two gilding metals of limiting grain size after cold work to 6-numbers hard

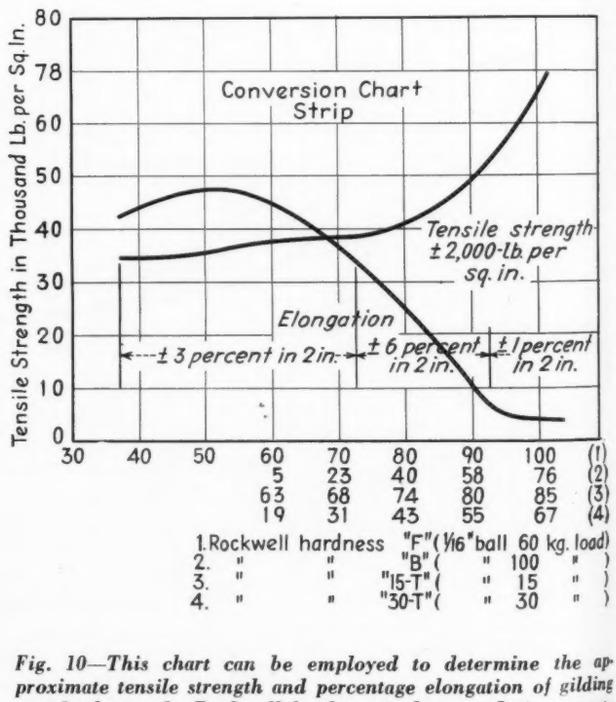

Conversion Chart Strip
Tensile strength ± 2,000-lb. per sq. in.
Elongation
± 3 percent in 2 in.
± 6 percent in 2 in.
± 1 percent in 2 in.
Tensile Strength in Thousand Lb. per Sq. In.
30 40 50 60 70 80 90 100 (1)
5 23 40 58 76 (2)
63 68 74 80 85 (3)
19 31 43 55 67 (4)
1. Rockwell hardness "F" (1/16" ball 60 kg. load)
2. " " "B" (" 100 ")
3. " " "15-T" (" 15 ")
4. " " "30-T" (" 30 ")

Fig. 10—This chart can be employed to determine the approximate tensile strength and percentage elongation of gilding metal when only Rockwell hardness is known. It is accurate for all thicknesses between 0.020 and 0.080 in. within the given limits. For example, when Rockwell hardness "B" is 40, the chart indicates a tensile strength of approximately 41,000 lb. per sq. in. and an elongation of approximately 25 percent

PHENOLIC RESIN COATINGS

Resinous coatings, which when baked are phenol-formaldehyde reaction plastic products of the thermo-setting type, because of their corrosion preventive properties make possible the use of coated parts made of steel or other materials to replace many of the critical materials in the corrosion-proof groups. In the viscous liquid condition as applied to the surface to be protected against corrosion, these coatings are solutions of phenolic resinoids. Application of heat causes the coating to harden by polymerization. When partially baked the coating is an insoluble elastic semi-solid. In this state another layer of liquid coating can be applied, after which another partial baking will fuse the layers together. After a thorough baking the resulting coating is a hard tough, permanently infusible and insoluble solid.

METHODS OF APPLYING COATING

(1) Heat should not be applied during coating operation. Method of applying coating will depend on factors such as quantity in production, size and shape of parts, and thickness of coat or extent of impregnation desired.

(2) Dipping Process may be done by hand or by mechanical device; parts that have no reentrant cavities or pockets which could entrap air can be dipped by machine.

(3) Spraying Process may be employed on parts that are too large to handle economically in a dip tank.

(4) Brushing Process should be used on pieces that have large flat surfaces. When brushing the coating should be applied rapidly, otherwise it may drag or become tacky. Coats should be drawn out well, with special attention to working the coating into all seams and crevices, and also to covering edges.

PROPERTIES OF BAKED COATING

(a) Hard, continuous, uniform composition
(b) Does not melt at any temperature
(c) Will not char at temperatures below 300 deg. F.
(d) Non-Hygroscopic, is unaffected by extremes of climate, temperature or humidity
(e) Does not deteriorate, is relatively unaffected by water, alcohol, acetone, benzine, common solvents, oils, greases, organic acids, dilute mineral acids, and common soap lubricants
(f) High mechanical bonding strength
(g) Good electrical insulator

Accelerated Tests on Phenol Coatings

Corrosive Agent	Percent Concentration	Time in days	Temperature in deg. F.	Effect
Acetic acid	5	1	240	Breakdown of coating
Carbon tetrachloride	100	8	170	None
Chromium plating bath	...	2	180	Breakdown of coating
Hydrochloric acid	10	8	214	None
Lactic acid	25	15	220	Slight color change
Sodium cyanide	10	1	240	Breakdown of coating
Sodium formate	10	12	220	Change in color
Stearic acid	100	11	183	None
Sulphuric acid	10	22	214	None
Sulphuric acid	10	4	227	None
Sulphurous acid	10	30	100	Breakdown of coating
White distilled vinegar	...	3	214	Breakdown of coating
Witch hazel	100	5	206	None

PREPARATION OF SURFACE FOR COATING

Surface should be dry and free from grease, rust and scale. Sandblasting or washing with grease remover is recommended. Rough or sharp edges should be smoothed or rounded off in order to secure good coverage at these areas.

DRYING OPERATION

After parts are dipped, sprayed or brushed, they should before final baking be placed or conveyed through a dryer oven to dispel solvents in the viscous coating. The drying operation prebakes the coating and partially polymerizes the resin. This initial baking requires about 15 min. at a temperature of 176 deg. F. for small work.

BAKING OPERATION

To completely polymerize the resinoid material in the coating the final coat is baked at higher temperatures. Satisfactory results are not likely to be obtained below 230 deg. F. For small work that can be placed in an

oven, baking at 248 deg. F. for 3 hrs., or 300 deg. F. for 1 hr. is usually required. Structures which are too large to place in an oven are baked by processes that employ portable equipment which can be moved to the job; this apparatus when set up provides a circulating blast of hot air to bake the coating. When the coating is to be applied to apparatus which is ordinarily operated at high temperatures, it is frequently possible to secure the baking effect by an application of heat in the vessel itself.

APPLICATIONS FOR HEAT-HARDENED COATINGS

Iron and steel parts wherever resistance to chemical reactions are required
Rayon and textile machine parts
Aluminum spools exposed to alkaline solutions
Textile spinning spools and paper-wound tubes
Centrifugal pump impellers to resist sulphuric acid
Builders hardware
Motor frames
Filter press linings
Flue liners in duct systems
Fume hoods
Fans, blowers and their housings
Battery cases
Heat exchangers
Steam jacketed kettles
Water softening tanks
Immersion heaters
Gas meters
Processing equipment subjected to the action of chemicals
Storage tanks
Shipping containers
Impregnating electro-magnet coils on electric chucks used on grinding machines to withstand the action of water, soap lubricants and other compounds
To seal joints on gasoline pumps
Coating for carburetor cork floats
Bonding agent for wood veneers and plywoods
Coating for wire trays
Pipe lines
Wooden fume stacks used in the phosphoric acid industry
Insulating bond and protective coating for coils, armatures and windings
Impregnating asbestos commutator rings, also for impregnating asbestos separating sheets between primary and secondary coils
Coating on horn fiber parts used in construction of cutouts

Resistance of Phenol-Formaldehyde Plastic Coatings to Corrosion

Corrosive Agent	Percent Concentration	Time in months	Effect
Brine with 1% H_2SO_4	...	24	None
Chromic acid	25	16	None
Hydrochloric acid	25	36	Change in color
Nitric acid	10	36	Change in color
Nitric acid	30	6	Breakdown of coating
Phosphoric acid	100	29	Change in color
Sulphuric acid	10	36	None
Sulphuric acid	50	36	Change in color
Sulphurous acid	15	20	None
Acetone	100	36	Slight color change
Acetic acid	5	18	None
Tannic acid in alcohol	5	18	Slight color change
Carbon tetrachloride	100	36	None
Citric acid	20	35	None
Ethyl alcohol	100	36	None
Formaldehyde	100	36	None
Formic acid	45	21	Slight color change
Gasoline standard	...	33	None
Lactic acid	100	30	Slight color change
Methyl alcohol	100	41	Change in color
Oxalic acid	8	18	Breakdown of coating
Carbolic acid	25	32	Change in color
Ammonium chloride	50	21	None
Ammonium fluoride acid	60	36	Slight color change
Ammonium fluoride neutral	60	36	Slight color change
Ammonium hydroxide	...	6	Breakdown of coating
Ferric chloride	10	12	Slight color change
Ferrous chloride	60	18	Change in color
Hypochlorites	3	13	Breakdown of coating
Sodium chloride	50	34	Change in color
Sodium hydroxide	2	2	Breakdown of coating
Aluminum sulphate	22	8	None
Ammonium alum	50	21	None
Ammonium sulphate	Saturated solution	18	None
Borax	50	21	None
Calcium chloride	5	19	None
Collodion salts	...	2	Breakdown of coating
Distilled water	...	30	None
Hydrogen peroxide	...	18	None
Magnesium carbonate	100	35	Slight color change
Magnesium hydroxide	23	35	Slight color change
Phosphorous trichloride	100	2	Change in color
Potassium alum	50	21	None
Potassium dichromate	5	35	None
Sodium bicarbonate	34	35	Change in color
Sodium bisulphate	29	36	None
Sodium chlorate	43	25	Change in color
Sodium cyanide	10	15	Change in color
Sodium fluoride	5	35	Change in color
Sodium hyposulphite	25	35	Slight color change
Sodium resinate	33⅓	21	None
Sodium sulphite	50	34	None
Sodium thiosulphate	50	16	Change in color
Zinc chloride	50	21	None
Zinc sulphate	...	32	None
Cotton seed oil	100	35	None
Glycerine	100	36	None
Kerosene	100	35	None
Malic acid	50	28	None
Napthenac acid	100	21	None
Oleic acid	100	5	None
Turpentine	100	35	None
Xylol	100	36	None

Data derived by Lithgow Corporation from tests on Lithcoted mild steel strips suspended in the various solutions at room temperature.

COATINGS OF SPRAYED POWDER

ALTHOUGH METAL SPRAYING with wire-guns for protecting metals with a thin coating of another metal or alloy has been available for some years, it was not until the powder spraying technique was recently introduced to this country by Schori Process Corporation that coatings of non-alloyable metals, mixtures of metals and non-metals, plastics, hard rubber, and even glass could be sprayed on metal, glass, plastics, wood, textiles and even paper. In the Schori process, finely powdered material is blown from an easily handled pistol through an oxy-acetylene or oxy-propane flame on to the base metal. The resultant coating is claimed to be less porous, has better adhesion, and can be applied faster than by previous metal spraying methods. Also, most materials for spraying are more readily available in powdered than in wire form.

Finely divided metal particles are blown onto the surface at a sintering or melting temperature, and apparently weld together with a strong mechanical bond between the metal coating and the roughened base surface. Because of the minute interstices between particles, a coating of sprayed metal built up to economically feasible thickness is usually slightly porous. Thicker coatings can be made dense and solid.

Thin sprayed coatings properly applied to sheet metal stand up well under forming, bending, and drawing operations, and do not chip off when pounded. Thicker coatings, however, are apt to flake off with mechanical abuse. Strength of the deposited metal film is good in compression, fair in tension, and poor in shear. It is this low shear strength that causes the coating to flake off rather easily if the mechanical bond between it and the base surface is destroyed.

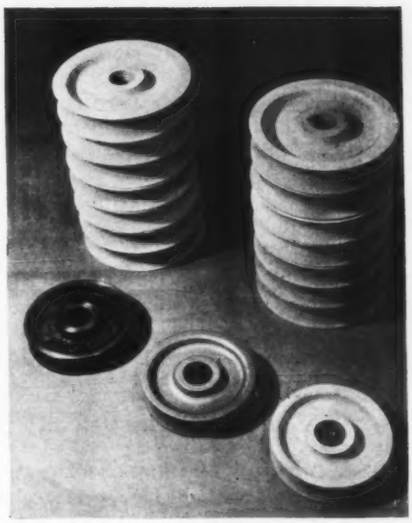
Surface Preparation

Blast cleaning of all metallic surfaces, to clean and roughen them, is essential. Tool roughening, wire brushing, and cyanide treatment, although sometimes successfully used, are not recommended.

With carborundum grit, size 30 or 36 grit is recommended. Angular steel grit, size 5 to 40, can also be used for cleaning ferrous materials. For sandblasting, Joplin grit is recommended. Air pressure should be not less than 80 lb. per sq. in. when cleaning steel. It can be lower with softer materials, with 65 lb. per sq. in. recommended for aluminum.

Parts which are covered with a film of oil should be de-greased before blast cleaning, either by vapor de-greasing or chlorinated solvent (trichlorethylene). Phosphate washes should not be applied to the prepared surface. They leave a poorly adherent film of salt.

No roughening of the surface is necessary before coating thermoplastics. The same is true of wood, but wood must be absolutely dry. Textiles to be coated should be dry and free from grease. When coating non-metals such as insulating materials, the surface should be clean and free from grease, and a thin coating of lead should be sprayed on first.



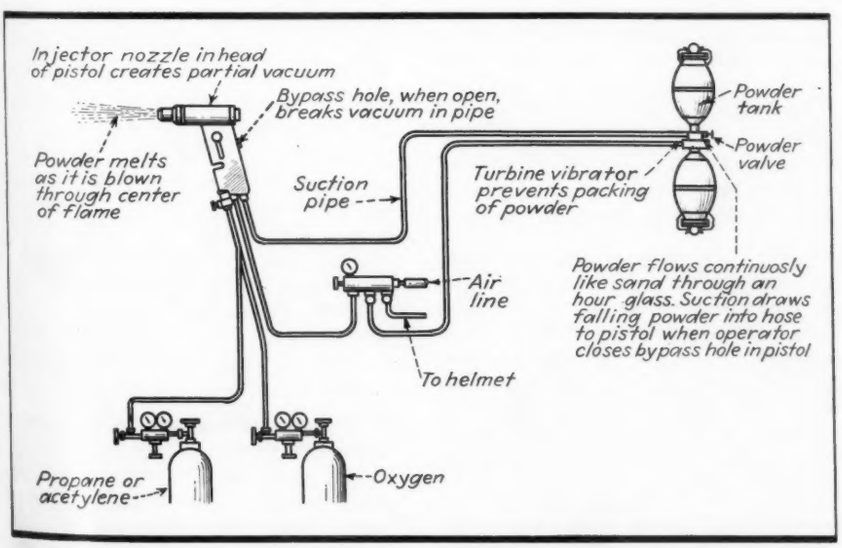
Malleable iron pulleys for life boats are sprayed with a coating of zinc for corrosion protection

Glass can be coated readily if it is sandblasted or ground. A light sandblast is sufficient where the coating is to be very thin or translucent. In Europe a good bond between sprayed metal and glass has been obtained by chemically cleaning the glass—an expensive process—or by heating the glass to a plastic state while spraying on the metal.

Spraying and Finishing

The metal should be sprayed on the prepared surface as soon after blast cleaning as possible. This is particularly true for ferrous materials which should be coated not more than eight hours after cleaning. In damp climates not more than three hours should elapse. Longer periods might permit the formation of a detrimental oxide film.

In coating metal surfaces, the powder pistol is held about 10 in. away from, and as nearly perpendicular to the surface as possible. Although it usually does not matter how hot the base metal becomes, some metals such as light gage aluminum alloy should not be


Injector nozzle in head of pistol creates partial vacuum
Bypass hole, when open, breaks vacuum in pipe
Powder tank
Powder valve
Powder melts as it is blown through center of flame
Suction pipe
Turbine vibrator prevents packing of powder
Air line
Powder flows continuosly like sand through an hour glass. Suction draws falling powder into hose to pistol when operator closes bypass hole in pistol
To helmet
Propane or acetylene
Oxygen

Equipment and operation of Schori process for spraying powdered metal

heated above 250 deg. F. Higher temperatures will increase susceptibility to inter-crystalline corrosion.

Sprayed powdered metal coatings can be buffed and polished, but the operation is expensive. Where polishing is required, the deposited coating should be made 0.012 in. thicker than required for the finished coating. Approximately five grinding and buffing steps are required.

Sprayed metal coatings are often followed by paint where greater corrosion resistance is desired. The matte finish made by the sprayed coating is an excellent key. The surface should first be primed with a phenolic resin base primer with zinc-chromate or zinc-tetraoxic-chromate pigment. The first coat of the primer should be diluted so as to flow into all the interstices between the minute metal particles.

Metallic Coatings

ZINC is perhaps the most important sprayed metal coating for protecting sheet metal, castings, forgings, and welded steel structures from atmospheric corrosion. It is particularly useful for patching galvanizing, and for zinc-coating heat-treated materials such as malleable iron which may warp in a high temperature galvanizing bath.

Thickness of zinc sprayed coatings should be from 0.004 to 0.006 in. to withstand mild corrosive atmospheric conditions. For medium severe atmospheric conditions, the same thickness of zinc coating is applied, followed by a zinc-chromate primer and paint. Where corrosive conditions are severe and the part is subject to some mechanical abuse, a coating of zinc 0.009 to 0.011 in. thick followed by primer and paint should be applied. Where corrosion is severe and there is no mechanical abuse, the zinc can be applied as thick as is economically feasible. Coatings as thick as $\frac{1}{16}$ in. have been applied to shafts in service under salt water.

ALUMINUM sprayed coatings are primarily for protecting steel and aluminum alloys. On steel they are usually from 0.004 to 0.006 in. thick. Slightly thinner aluminum coatings are recommended for spraying on aluminum alloys; heavier coatings may flake because the bond between aluminum and the sprayed coating is not as good as with steel. Where corrosion conditions are severe, the aluminum coating can be followed with zinc-chromate primer and paint.

ZINC-ALUMINUM. A coating of sprayed zinc from 0.009 to 0.011 in. thick followed by a coating of sprayed aluminum from 0.004 to 0.006 in. thick gives a corrosion-resistant finish which is exceptionally good for the service to which oil-tankers are subjected. Tanks sprayed with this double coating have long life although subjected to the corrosive action of salt water, fuel oil and gasoline.

HEAT-TREATED ALUMINUM. A sprayed metal coating superior to high-chrome steel in heat resistance is used quite extensively in Europe for light burners, engine manifolds and parts of industrial furnaces. In this process a sprayed coat of aluminum 0.004 to 0.006 in. thick is covered with bitumin or sodium-sulphate. The part is then heat-treated at 1,650 deg. F. for 20 min. After treatment the surface consists of films of various iron-aluminum alloys and their oxides. The life of this coating is greatly diminished at temperatures exceeding 1,750 deg. F. because of diffusion of the aluminum into the steel base. This method of heat-treatment is convenient especially for large articles which would be difficult to treat by the cementation processes in which packing a powder in heated drums is necessary.

LEAD coatings fall into two principal application classes: (1) Thick chemical-resistant coatings applied in thicknesses from $\frac{1}{32}$ to $\frac{1}{16}$ in. These thick coatings are non-porous, but cannot withstand mechanical abuse and are used principally as a substitute for lead burning. (2) Of increasing commercial importance as an atmospheric corrosion-resistant finish for steel, lead is sprayed to a thickness of 0.004 to 0.006 in. This is followed by zinc-chromate primer and paint, and is recommended as a substitute for galvanizing, electroplating, and aluminum coating.

Lead or lead bearing alloy has been sprayed on locomotive spring leaves as a lubricant. The thickness of these coating ranges from 0.009 to 0.011 in. Lead-antimony-silver alloy has also been sprayed on anodes to act as a catalytic agent in the electrolytic copper process. Another interesting application of sprayed lead alloy is on electrical porcelain to permit making solder connections directly to the porcelain parts.

NICKEL, CHROMIUM, STAINLESS STEEL. Sprayed coatings of these metals are not recommended for corrosion resistance, unless of adequate thickness. Thin coatings are porous, and these metals are on the wrong side of the electrolytic series when applied to steel. Very heavy coatings of nickel have successfully been applied for repairing nickel-clad chemical equipment.

COPPER, TIN, SILVER. These materials, also, are unsuitable in thin coatings for corrosion resistance. Copper has often been used in various types of electrical equipment. Wooden boxes for radios have been sprayed with copper for shielding, and copper has been sprayed on rubber to prevent accumulation of static electricity.

Organic Coatings

Organic coatings applied by the Schori process are non-porous. The melting points of the powdered materials are so low that the minute particles flow together over the heated surface. Non-porous coatings cannot be applied to materials such as paper and wood which cannot be heated above the melting temperature (approx. 250 deg. F. for Thiokol) of the material being applied.

THIOKOL. A sprayed coating of Thiokol $\frac{1}{32}$ in. thick is resistant to dilute acids and alkalis. Thiokol coatings become brittle below freezing, and become semi-plastic at temperatures above 140 deg. F. This synthetic rubber is resistant to petroleum and most organic solvents, and is particularly suitable for lining tanks and chemical equipment where the lining comes in contact with both acids or alkalis and petroleum or organic solvents. Other applications are for abrasion resistance. The natural color of Thiokol is a dirty green, but it can be pigmented with dark colors.

NATURAL RUBBER. Sprayed coatings of powdered hard vulcanized rubber will stand somewhat higher concentrations of alkalis and acids than synthetic rubber. The coating is very hard compared to that of Thiokol and is brittle at or below room temperature. Its high temperature limit is higher than that for Thiokol, natural rubber being suitable for temperatures between zero and 180 deg. F. The temperature resistance of hard rubber compounds vary inversely with their acid resistances.

CASHEW NUT PLASTICS. Cashew nut phenol-formaldehyde plastics can be sprayed, but at the present stage of development they are too brittle for most applications. The coating is of a dark brown color, does not have a high gloss, and has a wavy surface unsuitable where appearance is the primary objective. Principal application is for electrical insulation, such as on ignition wires for aircraft. Temperature range is from —10 to 130 deg. F.

GLASS COATINGS, acrylic resins, polymerized ethylene (polyphene), and cellulose acetate and butyrate coatings are all in process of development. Some degree of success has been attained with these materials, but they are not yet ready to be introduced commercially.

ANTIMONY—LEAD ALLOYS

Antimony, used as an alloying addition to lead, is a hardening and strengthening agent. High percentages of antimony make the alloy brittle but this can be reduced by the addition of tin. Small quantities of tin in antimony-lead alloys increase their stiffness but do not increase their brittleness as much as would a corresponding increase of antimony. Tin and antimony both increase the deformation pressure of either high-purity lead or a copper bearing lead to about the same value.

Metal Additions

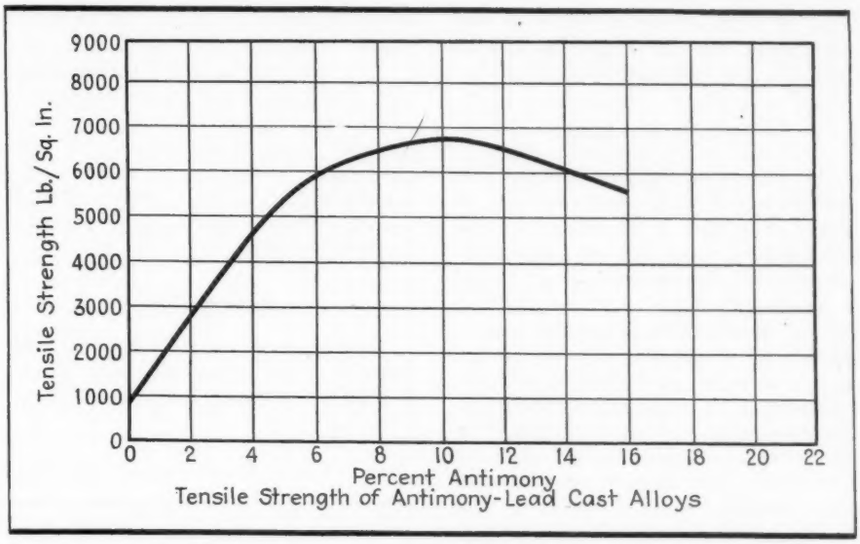
Tin, arsenic, copper, tellurium, nickel, bismuth and some other elements improve mechanical properties of antimony-lead such as hardness, creep strength, and fatigue strength and also stabilize these properties at temperatures above normal. Arsenic additions prevent segregation, raise strength and hardness at normal temperatures and help to stabilize properties at higher temperatures. Up to 0.10 percent of arsenic, however, when present as an impurity in dispersion-hardenable alloy has no effect in increasing hardness where the content of antimony is only one percent. Also small amounts of copper, bismuth, cadmium, nickel, and mercury, when present as impurities in bearing alloys containing 12.50 percent antimony, 3.00 percent arsenic, and 0.75 percent, do not affect the properties noticeably.

Copper hardens the antimony lead alloys and 1 to 2 percent copper will also tend to prevent segregation of tin and antimony when the tin content is appreciable. When tin content is low there is little tendency for such segregation.

Hardness

Physical properties of antimonial lead bearing alloys, at least as determined by hardness, are inferior to those of standard tin-base alloy. However, the antimony-lead bearing retains its hardness better than tin babbitt. Tensile strength data confirm the hardness data and show that while antimonial lead is initially slightly softer at room temperature than lead or tin-base babbitt, at more than 212 deg. F. it is very much better than these alloys.

Brinell hardness of a given composition is variable, depending upon temperature, cold working, age-hardening, and perhaps other variables.



Tensile strength of antimony lead alloy varies with the antimony content and reaches a maximum at about 10 percent antimony. Tensile strength also varies with heat treatment and changes during aging

Table I—Chemical Composition and Physical Properties

(Properties at 68 deg. F.)

Alloy No.	Antimony %	Tin %	Arsenic %	Lead %	Tensile Strength Lb/sq. in.	Elongation Percent	Brinell Hardness
1	5.0	....	....	95.00	6,900	8.00	9.2
2	10.0	....	....	90.00	8,000	6.00	13.5
3	12.0	....	....	88.00	9,300	5.25	14.5
4	12.5	0.65	0.65	86.20	9,800	5.50	18.0
5	12.5	0.75	3.00	83.75	10,200	2.80	22.0

Alloy 1 is used extensively for battery grids, storage battery terminal connectors, fire extinguisher nozzles and similar parts.

Alloy 5 retains hardness even after long periods at high temperature.

Table II—Physical Properties of Aluminum and Zinc Alloy Die Castings

Alloy	Tensile Strength	Elongation Percent	Brinell Hardness	Charpy Impact
Aluminum	29,000	1.0	...	1.5
	33,000	4.0	...	5.0
Zinc	40,300	3.0	60	17
	44,500	5.1	100	20

Tensile Strength

Tensile strength of a given antimony-lead alloy is also variable. Quenching from 480 deg. F. produces a tensile strength of about 2,800 lb. per sq. in. which subsequently increases to about 6,000 lb. per sq. in. as the result of aging at room temperature. If cooled slowly from 480 deg. F. the alloy does not undergo this wide variation in strength. In aging, the antimony migrates to the grain boundaries and results in porosity of the alloy.

Fatigue Strength

Fatigue strength of the antimony-leads is higher than that of conventional lead-base and tin-base babbitts. This is important in cable sheath as well as in bearings but because of de-

velopment of porosity in antimony-lead cable sheath attempts are being made to substitute calcium as the alloying metal. An alloy developed a dozen years ago for cable sheath, which contained 0.50 percent antimony and 0.25 percent cadmium, proved to be a great improvement over pure lead.

Cold Working

Lower members of the antimony lead series are soft in the cast condition but may be hardened by cold-work. The work-hardness is lost at amospheric temperature by self-annealing, but at a slower rate than with tin-containing alloys. Antimony-leads are also susceptible to age-hardening, but the hardening develops relatively slowly. Increase of antimony increases susceptibility to age hardening, even after cooling from the eutectic temperature at relatively slow rates, for example after casting. Cold working, however, destroys age-hardening effect, and softening is carried still further by self-annealing during storage at atmospheric temperature. An outstanding alloy in this group contains 1.5 percent cadmium and 0.5 percent antimony, which age hardens with extreme readiness, although it may not be permanent.

In alloys to be used in a severely cold-worked condition, such as cold-rolled sheet, little advantage accrues from the presence of more than a relatively small proportion of alloying addition. Cold-rolled 1 percent antimony-lead alloy is little harder than cold-rolled one percent antimony-lead alloy. Brinell hardness of cast antimony-lead alloys containing 5-12 percent of antimony is about the same as that of several other alloys of this class after age-hardening.

Alloys most susceptible to age hardening are those containing 2 to 3 percent antimony; 2 percent antimony and 1 percent cadmium; and 1.5 percent cadmium and 0.5 percent antimony. Age-hardening of the last named alloy takes place so readily that it is difficult to suppress the effect by slow cooling.

Even the hardest alloy is relatively soft. Pipe plugs of antimony-lead proved unsatisfactory because of inadequate strength and susceptibility to distortion. Tightening and loosening the plug with a wrench caused cold flow at the corners of the nut. On other parts the softness of the alloy prevents the use of thin sections except where no stresses or loading are involved.

Die Castings

Although low melting point of the antimony-lead alloys makes for easy casting and long die life, the low tensile strength and low impact strength of these alloys limits their use to applications where strength, lightness, and resistace to cold flow are not particular requirements. Extremely close dimensional limits can be maintained with lead alloy die castings and this is sometimes a factor in their selection. Lead alloys are highly resistant to corrosion and their high specific gravity also accounts for the use of some lead alloy die-castings. These alloys can be welded by gas torches and other suitable heating appliances and can be soldered readily.

Slush Molding

A slush molding technique is used when it is not required to meet specifications of strength, interchangeability, or stiffness. Such a casting is cheap, easily soldered, has a bright and comparatively non-tarnishing surface requiring no polishing operation, and can be made economically singly or in quantities with an inexpensive mold. Ornamental novelties are commonly made by this method. Difficult castings can be made in sections and assembled by using a soldering torch.

Finish and Electroplating

Lead base alloys can be painted or electroplated as readily as other metals and alloys. Electroplated lead base alloys are exceptionally resistant to severe salt spray tests, high humidity, and similar conditions. In this respect lead alloy die castings are superior to zinc when proper procedures are used.

Table III—Brinell Hardness as Related to Temperature

Composition, percent					Brinell Hardness					
						At 30 C.		At 150 C.		
Antimony	Arsenic	Tin	Copper	Lead	As Cast	After 7 days at 150 C.	After 46 days at 150 C.	After 17 hr.	After 7 days	After 46 days
12.65	2.97	0.65	....	Bal.	22	20.5	19.5	10	9.7	9.3
15.00	0.20	5	0.50	Bal.	20	16.5	15.5	7.3	7	5.8
13.89	0.34	9.42	00.44	Bal.	22.3	16.5	....	7	6.6	...
6.25	....	88.25	5.50	...	25	22.3	20.9	9.5	9.3	8.3

Table IV—Tensile Properties at Various Temperatures

Composition, percent					75 deg. F.		212 deg. F.		300 deg. F.		392 deg. F.	
Antimony	Arsenic	Tin	Copper	Lead	Tensile Strength Lb/sq. in.	Elongation % in 2 in.	Tensile Strength Lb/sq. in.	Elongation % in 2 in.	Tensile Strength Lb/sq. in.	Elongation % in 2 in.	Tensile Strength Lb/sq. in.	Elongation % in 2 in.
12.6	3	0.75	...	Bal.	9,800	1.5	6,700	4	4,200	10	1,900	70
15	0.2	5	0.5	Bal.	10,000	5	5,400	27	2,900	55	1,200	100

Table V—Cold Rolling and Aging Effects

Brinell Hardness Numbers

Antimony Percent	Before Rolling	After 50% Reduction	After 80% Reduction	After 80% Reduction and 10 days Aging at 32 F.
0.5	5.4	7.6	8.0	5.4
1.0	12.1	11.4	10.1	6.4
2.0	20.5	14.9	13.1	7.3
3.0	20.5	15.2	12.0	6.4

PROPERTIES OF PLYWOODS

FACTORS IN DESIGN OF PLYWOOD STRUCTURES

Engineering properties of plywood panels depend not only upon the physical properties of the specific species of woods used for the various plies or laminations, but also upon the number and thickness of the plies and the direction of the grain of the wood in the various plies.

A single veneer, depending upon the species, in general has a tensile strength 3 to 5 times the compressive strength for loads applied in the direction of the grain. Also the tensile strength along the grain may be 20 or more times the tensile strength perpendicular to the grain. In addition the modulus of elasticity along the grain is from 15 to 80 times that across the grain.

By gluing together plies in a manner such that the grain of the wood in one ply is at right angles to the grain of wood in the adjacent plies, plywood panels can be constructed that have about the same strength both parallel and perpendicular to the edge of the panel. Only plywoods of this type of construction are considered in this article.

Symmetrical stress distribution in plywood panels is obtained by using an odd number of plies, and by arranging the plies so that for any ply of a particular thickness there is a parallel ply of the same thickness and of the same species or of one having similar characteristics on the opposite side of the core equally removed from the core.

Where resistance to splitting is desired, as in plywood that is fastened along the edges, a large number of plies affords a stronger fastening than does a plywood made up of a few plies.

In panels of many thin plies the glued joints between plies will not be stressed by shrinkage strains caused by changes in moisture as much as would the joints in panels constructed of a smaller number of thicker plies.

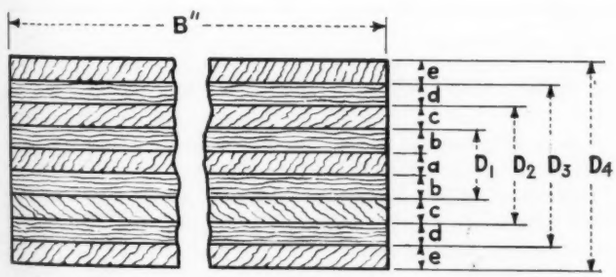
SIMPLE TENSION PARALLEL AND PERPENDICULAR TO GRAIN

The greater the number of plies used for a given panel thickness, the more nearly will the tensile strength parallel to the edge of the panel approach the strength perpendicular to the edge of the panel when the plies are alternately parallel and across grain. Because the strength of a single ply in the direction of the grain is so much greater than the strength across the grain, the strength of only the plies with grain parallel to the direction of load are considered.

BENDING AND DEFLECTION OF PLYWOOD PANELS

For beam structures the moment of inertia of the cross section is computed by considering only those plies that have their grain running in a direction parallel to the span. For horizontal panels with plies of equal thickness, therefore, since the number of plies having their grain parallel to the face plies is one greater than the number of plies with grain at right angles to the face, the moment of inertia is greatest when the grain of the face plies is parallel with the span. Data covering form factors used to calculate the strength of composite beams and columns should be arrived at after consultation with the plywood manufacturer.

(*Continued on next page*)



Cross Section of Plywood Panel with plane of plies at right angles to direction of load

D_1 = *Thickness of 3-ply panel in inches*

D_2 = *Thickness of 5-ply panel in inches*

D_3 = *Thickness of 7-ply panel in inches*

D_4 = *Thickness of 9-ply panel in inches*

Properties of Plywood Panels of Rectangular Cross-section

Moment of Inertia $I = (K_1B)/12$

Section Modulus $R = (K_1B)/(6K_2)$

GRAIN OF FACE PLIES PARALLEL TO SPAN

Number of plies in panel	K_1	K_2
3	$D^3_1 - a^3$	D_1
5	$D^3_2 - (a + 2b)^3 + a^3$	D_2
7	$D^3_3 - (a + 2b + 2c)^3 + (a + 2b)^3 - a^3$	D_3
9	$D^3_4 - (a + 2b + 2c + 2d)^3 + (a + 2b + 2c)^3 - (a + 2b)^3 + a^3$	D_4

GRAIN OF FACE PLIES AT RIGHT ANGLE TO SPAN

Number of plies in panel	K_1	K_2
3	a^3	a
5	$(a + 2b)^3 - a^3$	$(a + 2b)$
7	$(a + 2b + 2c)^3 - (a + 2b)^3 + a^3$	$(a + 2b + 2c)$
9	$(a + 2b + 2c + 2d)^3 - (a + 2b + 2c)^3 + (a + 2b)^3 - a^3$	$(a + 2b + 2c + 2d)$

PROPERTIES OF PLYWOODS *(Continued)*

Strength Values of Various Woods Parallel to Grain

Based on 15 percent moisture content

(From tables compiled by U. S. Forest Products Laboratory)

Species	Specific gravity average	Static Bending			Compression fiber stress at elastic limit, lb. per sq. in.
		Fiber stress at elastic limit, lb. per sq. in.	Modulus of rupture, lb. per sq. in.	Modulus of elasticity, 1,000 lb. per sq. in.	
HARDWOODS					
Ash, black	0.53	6,400	11,900	1,340	4,050
Ash, commercial white	0.62	8,900	14,800	1,460	5,250
Basswood	0.40	5,600	8,600	1,250	3,370
Beech	0.66	8,200	14,200	1,440	4,880
Birch	0.68	9,500	15,500	1,780	5,480
Cherry, black	0.53	8,500	12,500	1,330	5,100
Cottonwood	0.43	5,600	8,600	1,190	3,520
Elm, rock	0.66	7,900	15,000	1,340	5,180
Gum, red	0.53	7,500	11,600	1,290	4,050
Hickory	0.79	10,600	19,300	1,860	6,520
Mahogany, African	0.47	7,900	10,800	1,280	4,280
Mahogany, true	0.51	8,800	11,600	1,260	4,880
Maple, hard	0.67	9,500	15,000	1,600	5,620
Oak, white and red	0.69	7,800	13,800	1,490	4,950
Yellow poplar	0.43	6,000	9,100	1,300	3,750
SOFT WOODS					
Cedar, incense	0.36	6,000	8,700	1,020	4,320
Cedar, northern white	0.32	4,700	6,600	700	3,040
Cedar, Port Orford	0.44	7,400	11,000	1,520	4,880
Cedar, western red	0.34	5,100	7,800	1,030	4,000
Cypress, southern	0.48	7,100	10,500	1,270	4,960
Douglas fir	0.51	8,000	11,500	1,700	5,600
Pine, northern	0.38	5,900	8,700	1,140	3,840
Pine, Norway	0.51	8,500	11,900	1,560	5,280
Pine, sugar	0.38	5,600	8,000	1,040	3,680
Pine, western white	0.42	6,000	9,300	1,310	4,240
Spruce	0.40	6,200	9,400	1,300	4,000

Modulus of rupture is not a true stress, it is the value obtained by substituting maximum bending moments of rectangular beams in the ordinary beam formula. When the design stresses for fiber stress at elastic limit in bending, or the modulus of rupture, as given in the Table are used in the design of beams the sections of which are not rectangular, factors whose values depend upon the shape of the cross section must be applied.

SPECIES OF WOOD FOR FACE PLIES TO MEET SERVICE REQUIREMENTS

Woods checked in column *A* are used to obtain hardness, resistance to abrasion and fastening strength.

Woods checked in column *B* are used where finish or appearance is required.

Woods checked in column *C* are used for plywood that must be steamed or soaked, before being bent into a form or shape that is to be permanent after forming.

Woods checked in column *D* are used for flat panels in which high bending strength, or high column strength, with minimum weight is required.

Species	*A*	*B*	*C*	*D*
Beech	√	√	√	
Birch, sweet or yellow	√	√	√	
Hard maple	√	√	√	
Black walnut	√	√	√	
Soft elm		√	√	
Red gum, heart		√	√	
Soft maple		√	√	
Mahogany, African or true		√	√	
Sycamore		√	√	
Basswoods				√

Species	*A*	*B*	*C*	*D*
Port Orford cedar				√
Spanish cedar				√
Fir, low land white, noble or silver				√
Tupelo gum				√
West Coast hemlock				√
Pine, white, sugar, pondosa, California white				√
Yellow poplar				√
Redwood				√
Spruce, red, white or Sitka				√

Question and Comment

Factor for Determining Shell Thickness Of Copper Unfired Pressure Vessels

To the Editor:

The article on "Unfired Pressure Vessels" by Edward L. Maguire, in PRODUCT ENGINEERING for July 1941, pages 391-393, can be useful to any design engineer who has much tank design work. I was glad to see the correction notice published on page 441 of the August number. The obvious error in transposing the subscripts for the scales on page 393 tripped us up for a while when we checked the reference sheets for accuracy.

The chart, page 391, checks with the A.S.M.E. Unfired Pressure Vessels Code and Paragraph U-122. Also, the chart on page 392 checks with Paragraph U-20.

The chart for "Shell Thickness with Internal Pressure" on page 392 would be of greater use to us if we were able to apply it for hard and soft copper. In Note (3) at the foot of the page you list factors by which the t of stainless steel is to be multiplied to determine shell thickness with other metals. Have you similar factors for hard and soft copper? —FRED W. STRADTMAN
Chief Draftsman
Rice & Adams Corp.

To the Editor:

In reply to Mr. Fred W. Stradtman's inquiry regarding my article on "Unfired Pressure Vessels", factors for determining shell thickness from the chart on page 392 can easily be worked out for any metals by referring to the A.S.M.E. Code for Unfired Pressure Vessels.

The equations for determining shell and dished head thickness for a given application would have only one variable—the tensile stress of the materials from which the tank is to be made. A factor by which the charts can be applied to any material can therefore be determined by selecting the allowable stress at the temperature at which the vessel will operate, as given in the table for non-ferrous metals (Table U-3 in the A.S.M.E. Code), and multiplying this value by five. The resultant figure is then divided into 75,000 (the minimum ultimate tensile stress of stainless steel). Factors thus determined can be multiplied by the t of stainless steel as determined from the chart to give an accurate value for the t of a vessel made from any material. —E. L. MAGUIRE

[*Editor's Note*—We have applied Mr. Maguire's factor calculation to the copper tubes, plates and pipes as listed in Table U-3 of the A.S.M.E. Code. This table lists an allowable stress of 6,000 lb. per sq. in. for copper at temperatures up to 150 deg. F. The allowable stress is only 4,000 lb. per sq. in. at a temperature of 400 deg. F., and the material is not recommended for higher temperatures. Factors for these temperatures are computed as 2.5 for 150 deg. F., and 3.75 for 400 deg. F.

Note that these factors cannot compare with the factors listed for other materials on page 392 which are computed on the basis of allowable stress at 700 deg. F.]

U. S. Army Knows German Artificial Fog

To the Editor:

In your "Engineering in Washington" column for August, page 445, under the sub-head "Artificial Fog," you state that you could not find anyone who had known of artificial fog being used on land.

In talks with Germans employed by American Cutting Alloys, Inc., early in 1937, statements were made by them that the German army was then experimenting with an artificial fog for land use.

Their statements were that the fog produced was of light gray or almost white color, very similar to natural fog; that it could not be penetrated by any known light; that it was harmless, except when small quantities of mustard gas was used with it; and that the addition of mustard gas gave the fog a yellowish color.

As I understand the statements, this artificial fog could not have been titanium-tetrachloride as that gas has a black or deep brown color—not white.

Statements were also made that this fog could be drifted from airplanes over land or water to hide their operations, that it diffused evenly in the air as it settled toward the earth or toward the water, that it settled so slowly that airplanes could operate for nearly a half an hour behind a bank of fog.

If the Army officials have stated that they knew nothing of this, there must be a mistake somewhere, for I spoke of it when visiting Colonel Jenks with Mr. P. T. Wetter late in 1937 or early in 1938. It would seem that the composition of the gas was a state secret, as the engineers did not refer to the type of chemicals used.

—C. G. WILLIAMS, M.E.
Davenport, Iowa

Half Truths More Harmful Than Lies to Defense

To the Editor:

I wish to commend you both for your editorial "Play Hard to Win" and your article "U. S. Armament Superior" in August PRODUCT ENGINEERING. Both deserve nationwide publicity.

I particularly endorse your remarks about the "warmongering" Senatorial investigation. It is surprising how some people who think they are intelligent believe the allegations of the man who headed up this investigation. He seldom tells both sides of a story. In my estimation, a half-truth is worse than a lie.

At this college we have been trying to do our part by painting the true picture of our economic system. Every engineer graduating here, knows what industry is doing for him and everyone else in this nation.

In our machine design classes, we have used ordnance problems for the last ten years in order to make our graduates more useful to both Army and industry. Many of them are now in Army, Navy or Marine Corps doing valuable work.—CARROLL D. BILLMYER
Assistant Professor
Rhode Island State College

[*Editor's Note*—We thoroughly agree that a half-truth is worse than a lie, but the worst of it is there are even more half-truths than lies, and there are

plenty of the latter. When magazines and newspapers, which are presumably of the highest character, stoop to some of the tales which are now appearing, it is really frightening.]

Wrinkle Finishes Repaired By Simple Retouching

To the Editor:

Here is a question to which one of your readers may have an answer. Is there any satisfactory method for covering up scratches on wrinkle finished surfaces?

We have tried out various ideas, such as holding an infra-red lamp close to the surface for a period of time after applying the finish. Nothing to date has been very successful.

—E. W. PETERSON
Chief Engineer
Ditto, Incorporated

To the Editor:

The way to retouch these scratches is to use an air drying material which is available from Ault & Wiborg Corporation and possibly from other manufacturers. More recently, with the introduction of infra-red drying, this retouching can be done successfully in some cases by retouching with the original finishing material and then baking it under the infra-red lamp. In this case, as in any infra-red finish operation, bulk of metal involved, the extent to which infra-red rays can be applied directly, and other factors must be considered. —DAVID DONOVAN
Interchemical Corporation

Mess Kits of Plastic

To the Editor:

When I was in the Army, mess kits were made of aluminum and probably still are. Why not make plastic substitutes and salvage the aluminum in the present mess kits?

—T. J. MCCARRON
New York, N. Y.

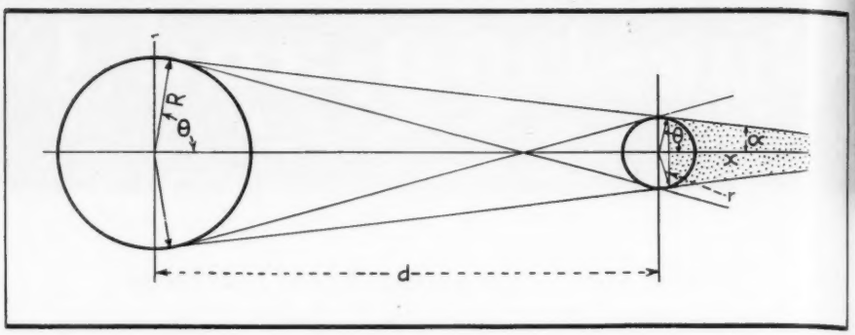
Equinoxial Problem

To the Editor:

This interesting problem is as far as I know, original. Although I have given the solution here, try to solve it without looking at the answer.

Problem: There are two days in each year when the duration of day and night are very nearly equal. These days occur when the earth passes through the vernal and autumnal equinoxes. Using the simplified values given below, determine how much longer the day is than the night when the earth passes through equinox.

Sun's diameter is 80,000 miles; Earth's diameter is 8,000 miles; and mean distance between centers is 93,000,000 miles.



Solution:

$$\frac{r}{x} = \frac{R}{d+x} = \operatorname{Sin} x$$

$$x = \frac{rd}{R-r}$$

$$\operatorname{Sin}\alpha = \frac{R-r}{d} \qquad \theta = 90 - \alpha$$

$$\theta = 90 - \operatorname{Sin}^{-1}\frac{R-r}{d} =$$

$$90 - \operatorname{Sin}^{-1}\frac{400{,}000 - 4{,}000}{93{,}000{,}000} =$$

$$90 - \operatorname{Sin}^{-1} 0.0042 = 90° - 14' 24'' = 89° 45' 36''$$

$2\theta = 179° 31' 12''$ in darkness =

$$179 + \frac{31}{60} + \frac{12}{60 \times 60} = 179.52°$$

$360 - 179.52 = 180.48°$ in light

$$t = \frac{24 \times 180.48}{360} = 12.03 \text{ hr.} =$$

12 hr. 1 min. 48 sec. light

Answer: At equinox there will be 3 min. 36 sec. (solar time) more light than darkness. —ROLAND W. ZIEGE
Central Scientific Co.

Can You Work This One?

H. E. SMITH

Solution to September problem—

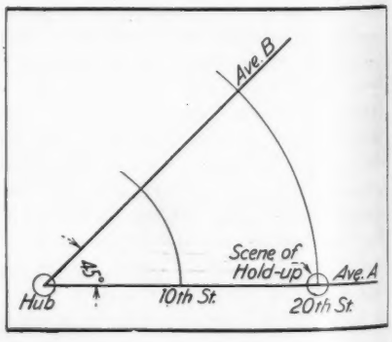
Tellers' Mistake

A man cashes a check, and by mistake the teller gave him as many cents as the check calls for dollars, and as many dollars as the check calls for cents. After spending $1.93, he finds he still has twice as much as the check was written for. By setting up equations, letting x-cents and y-dollars, it can easily be found that the check read $1.04.

This month's problem—

Cops and Robbers

In one section of a city, the avenues radiate at 45-deg. angles from a central hub, and the streets are concentric around the hub as shown, spaced 10 blocks per mile. One day a police car at the hub is ordered to rush to the scene of a hold-up at the intersection of 20th St. and Ave. *A*, and is told that the bandits are just making their get-away. The police car immediately starts driving out Ave. *A* at 30 m.p.h. through the heavy traffic. Three minutes later the police car is radioed that the bandits' car has just run through a red light at the intersection of Ave. *B* and 20th St., and is turning into Ave. *B* toward the hub. The police car immediately turns around, drives back along Ave. *A*, cuts through one of the streets, and arrives at Ave. *B* just in time to intercept the bandits. Assume constant speed of each car with no time lost in communication, and no time lost in stopping and turning. Through what street must the police car cross from Ave. *A* to Ave. *B* in order to just intercept the bandits?



News and Summaries

Experimental All-Plastic Ford Climaxes 12 Years of Research

ALTHOUGH IT MAKES ITS PREVIEW at a time when steel shortages threaten the auto industry's non-defense production, the all-plastic automobile body introduced unexpectedly in August by the Ford Motor Company is only a laboratory specimen, and the large-scale substitution of plastic for steel is still some years away, Ford officials hasten to explain. Problems still to be met are: inadequate supply of raw materials, molding capacity, and the lack of complete data on how the plastic body panels will stand up under weathering and unusual service conditions.

In appearance the "Dearborn" plastic car is not radically different from the conventional streamlined auto, with the exception that there are fewer deep drafts. Its frame is of welded steel tubing, on which are mounted the fourteen plastic body panels. Panels are attached by steel straps, as shown in Fig. 1. Where two panels join, they are cemented together with cold-setting phenolic resin. All molding and trim is of extruded plastic, attached by conventional methods. Windows are of acrylic plastic, the same material used for bomber blisters and windows.

Panels of the body are only ¼ in. thick, but they will absorb greater blows than steel panels without denting seriously. The total weight of the plastic panels is 250 lb., and the metal frame has a similar weight. The car with plastic body weighs about 2,000 lb.—a saving of 1,000 lb. as compared with a steel car.

The plastic developed in the Ford Research Laboratories is made from ordinary farm crops. Synthetic resins are mixed with fibrous filler materials derived from such crops as wheat, flax, ramie, hemp, and spruce pulp. To solve the problem of completely filling a large-area plastic mold, the plastic is "preformed" by suction into a sheet nearly the shape of the finished panel and about three times as thick. (See Fig. 3.) The preformed sheet is then inserted into a multi-tier press and molded at high temperature under 1,500 lb. per sq. in. pressure.

The value of preforming is twofold: (1) It permits a higher proportion of plentiful filler materials like paper pulp, slash pine fibers, soybean cellulose, ramie, etc., to the more scarce and expensive synthetic resins (one Ford formula is said to call for 70 percent filler—consisting of five parts southern slash pine fiber, 3 parts straw, 1 part hemp, and 1 part ramie—and only 30 percent of synthetic resin); (2) it obviates the cracking and pulling apart of high-filler plastic on deep draws and sharp bends.

"Plastic raw materials may cost a little more," Henry Ford admitted when asked about the economical aspects,

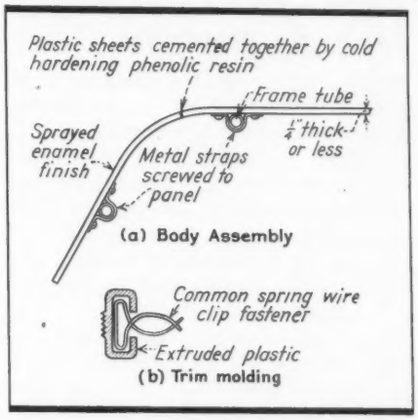

Plastic sheets cemented together by cold hardening phenolic resin
Frame tube
Sprayed enamel finish
Metal straps screwed to panel
¼" thick or less
(a) Body Assembly
Common spring wire clip fastener
Extruded plastic
(b) Trim molding

Fig. 1—Methods for mounting plastic panels and molding to frame of body

Fig. 2—Robert A. Boyer, who started at the age of 20 as research head of Ford's plastic car project, is shown here with one-quarter scale models of the automobile with plastic body and the welded tubular steel frame on which the panels were mounted. Fig. 3—In the preforming operation, a mixture of synthetic resin and fiber is sucked into a mold the shape of the finished panel

"but we anticipate a considerable saving as a result of fewer fabricating and finishing operations. For example, the relatively simple rear compartment door when made of steel requires no less than seven stamping operations, while only two are required for the same panel made of plastic."

Census Bureau to Survey Strategic Materials

DEFENSE AGENCIES of the Federal Government are asking the cooperation of American industry in ferreting out the quantities and whereabouts of strategic metals. The first of 65,000 questionnaires, addressed to the manufacturers using defense metals, have been put into the mail by the Bureau of Census, acting as agent for the O.P.M. and the O.P.A.C.S.

The questionnaires are simple in form, and ask the manufacturer to report the quantities on hand and uses made of antimony, cadmium, chromium, cobalt, copper, ferro-alloys, iridium, lead, manganese or spiegeleisen, mercury, molybdenum, non-ferrous alloys, tin, vanadium, tungsten, zinc, and scrap metal containing any of these. Results from these questionnaires will be tabulated by the Census Bureau and will be used directly for administrative check-ups.

When this information is compiled with reports from the Bureau of Foreign and Domestic Commerce and from the Bureau of Mines, an overall picture on the use, stocks and distribution of 90 percent of the listed metals will be available.

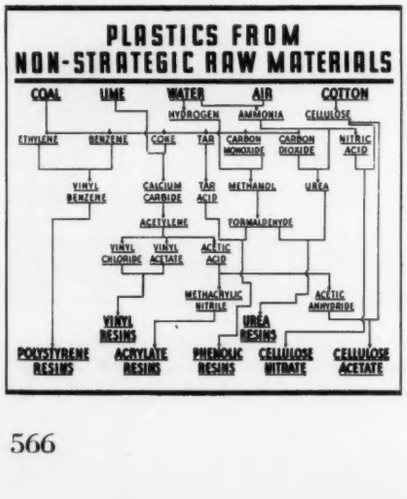
Most Plastics Unavailable Except for Defense

WITH DEMAND FOR PLASTICS rising, and raw materials being diverted to other defense purposes, there are practically no plastics available at the present writing for uses other than defense. All plastics using formaldehyde are now under mandatory control. Plasticizers for cast resins are rapidly becoming unavailable, and this fact may lead to serious restrictions in the supply of cellulose acetate and cellulose acetate-butyrate resins. The vinyl resins are unavailable except for defense purposes. Although there are now three manufacturers producing polystyrene resins, demand is way ahead of the supply. Cellulose nitrate, of relative unimportance, is still available.

Plastics authorities believe that all plastic materials may be placed under mandatory control within the next few months. When asked by PRODUCT ENGINEERING editors for a list of the order of relative availability for necessary applications, the following was given, with the most available at the top:

Cellulose nitrate
Polystyrene
Cellulose acetate
Cellulose acetate-butyrate
Vinyl resin
Acrylate resin
Phenolic resin
Urea resin

A study of the diagram at left shows why most of the shortages have occurred. Many of the intermediate materials, such as methanol, formaldehyde, nitric acid, and the plasticizers have been diverted to other defense uses.



York Plan of pooling men and machines for cooperative work on large government defense contracts is shown here in action as executives of York industries put their heads together over a $2,000,000 order for 6-in. Barbette guns. Left to right: P. P. Stock, methods, York Ice Machinery Corp.; H. C. Bentzel, methods, York Ice Machinery; Burwell B. Smith, vice president, S. Morgan Smith Co.; Warren C. Bulette, president, Brandt-Warner Mfg. Co.; William S. Shipley, chairman of board, York Ice Machinery; Marshall G. Munce, assistant to president, York Ice Machinery; Beauchamp Smith, vice president, S. Morgan Smith Co.; D. C. DeVine, general superintendent, S. Morgan Smith Co.; T. C. Cottrell, sales engineer, York Ice Machinery; M. L. Gotwalt, estimator, Read Machinery Co.; and Edward Fisher, works manager, A. B. Farquhar Co. Barbette guns are needed for harbor defense.

Hobart Offers Prizes For Arc Welding News

CASH PRIZES totalling $200 or more every month are now being offered for interesting news items about arc welding applications in a contest being sponsored by Hobart Brothers, Troy, Ohio. The prizes range from $100 for the first prize, down to additional special awards of $1 to $10 for entries deemed worthy of purchase by the contest judges. According to Hobart, no lengthy essay or technical data are required of contest entrants.

Do You Know That—

CHROMIUM-PLATED PLASTIC molding for automobile radiator grilles is in the experimental stage. Besides appearance and hard surface, chromium plating adds a slight amount of stiffness. Problems: to secure adequate bond between plated metal and plastic, to secure proper means for fastening, and to allow for slight expansion and contraction. (60)

TAPERED ROLLER BEARINGS submerged in a creek near Ichang, China, were retrieved after six weeks. After soaking in kerosene and refinishing, they were successfully used. (61)

ENGINEERING IN WASHINGTON

McGRAW-HILL WASHINGTON BUREAU, *PAUL WOOTON, CHIEF*

A floating drydock "about a block long" is being towed from shipyard to shipyard in England, relieving the most severe repair situations. It is moved by three ocean-going tugs and defended by destroyers, balloons and fighter aircraft, with a crew of 114 aboard.

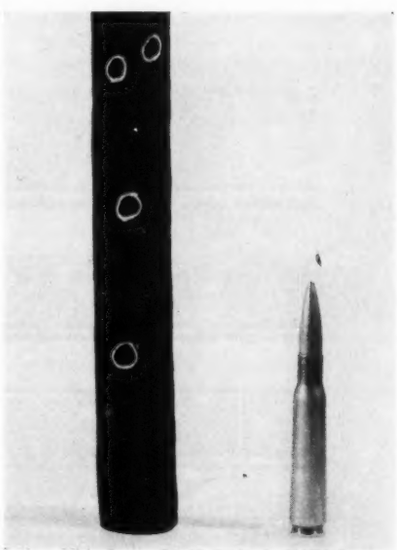
Winds eddy and flow around all kinds of structures, and they cause unsteadiness of aircraft operating near them. At Lakehurst, N. J., the Navy made preliminary tests with models of additional buildings to be put up. Smoke was blown over the models, from wind tunnels, and photographs of the currents were made.

In the first attempt, since the last war, to use concrete in shipbuilding, the Maritime Commission decides to build 100 reinforced concrete oil barges of five to eight thousand long tons capacity. The barges will require about half as much steel as for similar capacity in steel construction. And besides, reinforcing bars are less scarce than plates. Bidders are to submit designs. Three types are to be built. Two are conventional ship shape, one with screw and rudder, the other without either. The third is cigar-shaped—oval cross-section with conical ends. None is powered. In all types, maximum length is 370 ft.; maximum draft, 28.5 ft. Minimum slab thickness is 4 in., with at least ¾ in. cover for steel. Some concrete men say the Commission is acting moss-back in refusing to adopt concrete processes developed since the World War.

Army forks and spoons have been redesigned by the Quartermaster to eliminate zinc and nickel. New designs call for plated steel. Just to put an element of mystery in this page, we won't tell you why the knife wasn't done over. We don't know.

Two major improvements are being made on medium tanks. One is a power-operated turret, already standard on aircraft, and a new "gyro-stabilizer." Not much is being said about the stabilizer, but apparently it is an independently sprung gun mount held steady by gyroscopic action for steady firing while the tank lurches. At present, all tanks come to a stop for any accurate shooting. Ordnance is supposed also to have developed a "predictor" with which guns can be aimed while the tank is in motion. We still have no "heavy" tanks in production—only lights and mediums. The heavy model is still experimental; Baldwin Locomotive expected to finish a pilot model in a month or so at this writing.

To save 10,000 rounds of ammunition, in training each anti-aircraft machine gunner, the Coast Artillery has developed a device which saves 60 to 75 percent of training time. It is used on 30 and 50 caliber AA guns. Previously, thousands of rounds were wasted in firing at small balloons and sleeve targets before the men got the feel of handwheels. The details are too numerous to set down here, but in a word, the device consists of a 2½ x 9 ft. box in which the apparent trajectory of the tracer bullets is produced, by means of small electric lights, and a model target plane, operated by hand wheels.



Punctured by .50-cal. machine gun bullets, this bullet-sealing hose still held fuel under 10 to 15 lb. per sq. in. pressure without leaking. Developed by Goodrich, the hose has wall thickness of 5/16 in. with inner surface of oil-resistant Ameripol. Available in diameters from ½ to 3 in. An outgrowth of self-sealing fuel tanks, this hose will eliminate auxiliary metal fuel lines formerly needed in war planes.

One Standard gasoline will replace the four varieties formerly in use for Army vehicles in the field. The standard is 80 octane, which gives good mileage and is expected to cut down volume, which will please Mr. Ickes. Diesel fuels will be of three kinds instead of four; diesel lubricating oils the same; gasoline engine lubricating oil is cut from 10 types to three. Airplanes and medium tanks (which use airplane engines) will continue to use aviation gasoline which meets Army-Navy specifications.

Laboratory equipment and supplies have been given a priority rating of A-2. The National Academy of Sciences will advise in this new Research Laboratories Supplies Plan, concerning applications for rating or assistance. Any lab experiencing difficulty should apply to the Chemical Branch, OPM, Washington, on Form PD-88. Number of copies of the order necessary to serve upon the laboratory's suppliers, should be specified. If difficulty is experienced under A-2, application should be filed on Form PD-1, and special action will be taken if the project is sufficiently important to defense.

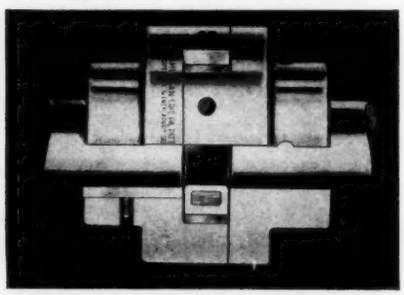
Use of wood by the Army will be increased under a new Wood Products Unit, under the Chemical Technical Unit of the Standardization Branch of the Supply Division of the Quartermaster. You can forget all those bureau names, we hope, and note that a lot of things like table legs, chests, etc., will be made of wood instead of metal.

New gas storage tanks are being studied by the Army. The 750 gallon gasoline tank mounted on the 2½-ton truck is not thought adequate for transport purposes. One type being tried out is a 2,000-gallon frameless tank on a semi-trailer. This container is re-inforced inside. The tractor would take the tank to the desired location, unhook and leave it to be unloaded. Another is a 2,000-gallon tank on a spring mounting for less wear. A third type is a 2,000-gallon self-sealing tank mounted on a semi-trailer. Material is canvas covered synthetic sponge rubber that seals itself after being punctured. The War Department refrains from saying they expect gunfire to spill some gas.

New Materials and Parts

Flexible Couplings

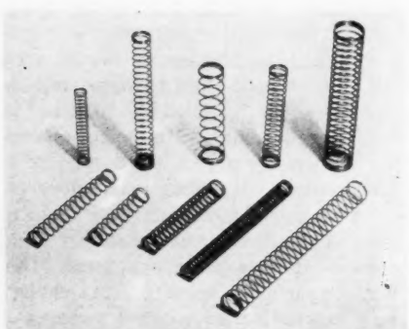
Designed for mechanical power transmission applications where higher torque capacity is required, two new series of heavy-duty American flexible couplings are now available. The Series *B* and Series *C* flexible couplings are basically similar to the Series *A* Amer-



ican flexible coupling, but the Series *B* has all-steel construction with hard chrome faces on the jaw flanges for heavy-duty installations. The Series *C* is of all-steel construction having steel jaw flanges and center member, together with Oilite bearing strips and hard chrome flanges. American Flexible Coupling Co., Erie, Pa.

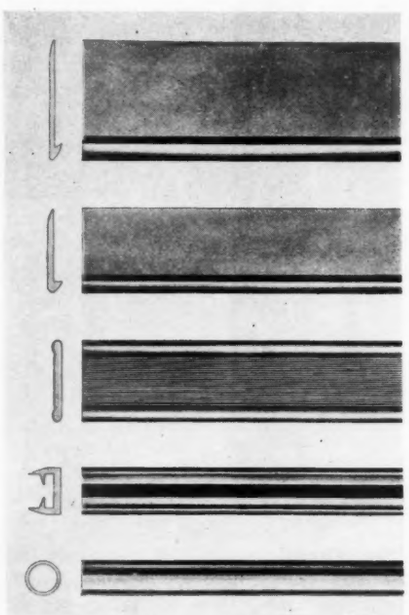
Beryllium-Copper Springs

Developed for small electrical applications, and made on special coiling equipment, these beryllium-copper springs have higher safe operating temperatures than bronze, music wire or stainless springs, and have more than twice the current-carrying capacity of bronze. The springs are supplied to the following tolerances in production quantities: length $\pm \frac{1}{32}$ in.; outside diameter ± 0.003 in.; load deflection, 5 percent. Springs are uniform in pitch and can be compressed to within $\frac{1}{8}$ in. of solid length without coils touching. Higher safe working stresses allow the springs to be designed for greater free length, thus reducing the change in brush pressure with brush wear. Being heat-treated for minimum drift, these brush springs will maintain initial assembled tension without set. High conductivity results from special heat-treating routine. Instrument Specialties Co., Inc., Little Falls, N. J.



Extruded Polystyrene

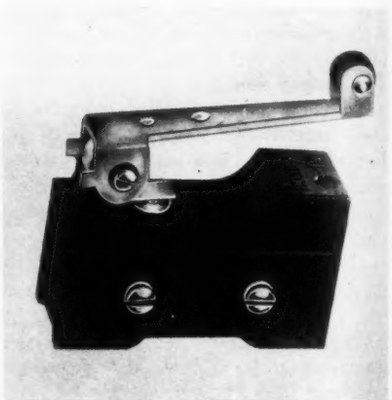
Difficulties in extruding polystyrene plastics have now been overcome, and tubing and special shapes of Lustron polystyrene are now available for the first time. The new process combines the advantages of Lustron's dimensional stability and superior electrical properties with the advantages of extruded shapes. Tubing can be made to specifi-



cations in diameters ranging from $\frac{3}{8}$ to 1 in.; special shapes can be furnished to specifications up to 4 in. wide. A drawing of the required cross-section, showing allowable tolerances, should be submitted in asking for quotations. Pierce Plastics, Inc., Bay City, Mich.

Roller Arm Actuator For Mu-Switches

Type Y actuator for a.c. and d.c. Mu-Switches combines the advantages of $\frac{3}{8}$ in. over-travel, large reduction in effective operating pressure, and construction suited to high speed cam operation, its nickeled steel arm carrying at one end a graphite impregnated, self-lubricating Bakelite roller and pivoted on an oil-less bearing at the other end. A U-spring of phosphor bronze, riveted to the under side of the arm, transmits the actuating force to the operating button of the switch and absorbs excess movement at the roller. Mu-Switch Corp., Canton, Mass.



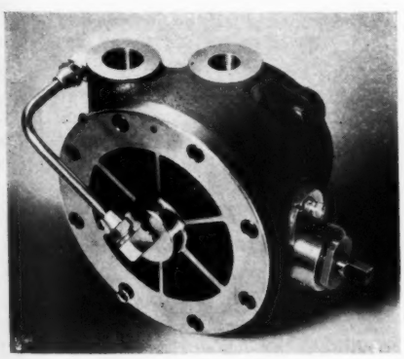
Whiteprint Machine

Fast printing speeds up to 12½ ft. per min. and uniform exposure essential to perfect printing are assured by a single high pressure mercury vapor lamp in the new Model *C* high speed automatic whiteprint machine. In the Model *C*, the functions of printing and developing are combined into one compact machine which can easily be installed in any office. The operator feeds the original and Ozalid sensitized material into the front of the machine where continuous belts carry both to the printer. After exposure the original is automatically separated from the print and the original is returned to the operator at the front of the machine. The print is then conveyed automatically to the developer, where after dry development, it is deposited either to a

delivery tray in the front of the machine, or to a sorting table at the rear of the machine. The Model *C* accommodates materials up to 42 in. wide. Ozalid Products Div., General Aniline & Film Corp., Johnson City, N. Y.

Hydraulic Pumps

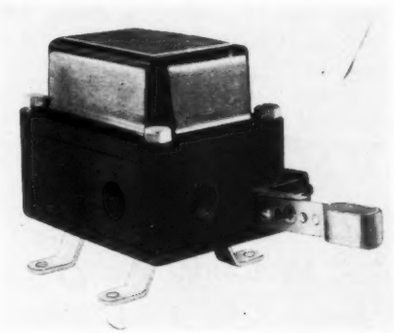
Infinitely variable delivery hydraulic pump is designed with a means of balancing the dual vanes so that overheating is eliminated, wear is reduced and the pump is virtually breakproof. Known as the "Dudco" pump and designed for continuous operation at high pressures, the new pump is recommended where the possibility of line



surges require a break-proof pump. Delivery is infinitely variable over two ranges: 4 to 0 g.p.m., and 8 to 0 g.p.m. The pump is recommended for continuous duty operation at 1,250 lb. per sq. in., but it can be used where periodic pressures go as high as 2,000 lb. per sq. in. Instead of the conventional single blade vane, the new Dudco pump has dual vanes. Each vane is beveled so that oil from the hole in the rotor can flow around all the edges of the dual vane. The result is that a back pressure, variable to suit individual conditions, can be placed on the vanes from the stator side. Thus, each set of vanes is held against the stator with only a portion of the total operating pressure of the pump, eliminating possibility of overheated vanes. Detroit Universal Duplicator Co., 253 St. Aubin, Detroit, Mich.

Float Switches

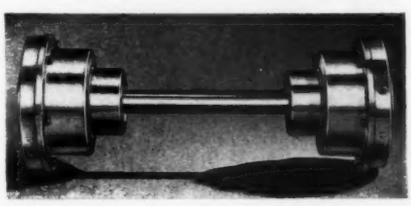
New type explosion-proof and water-tight float switches are inclosed in heavy cast iron covers and bases. In the Type 9048AW water-tight switch, shown here, a rubber gasket is provided to make it water-tight. The explosion-resisting float switch is identical to the 9048AW switch except that the surfaces between the cast iron cover and base are carefully machined to comply with Class 1, Group D hazardous location



requirements. Entries to the switch chamber are through approved openings by means of closely fitted steel shafts. Switches are available for circuits ranging from 110 to 550 volts a.c. or d.c. Square D Co., 6060 Rivard St., Detroit, Mich.

Piloted Coupling

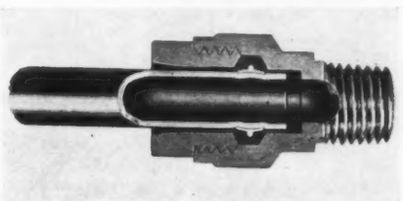
Especially adaptable for use in paper mills, on steel mill auxiliary devices, on main rolling mill drives, horizontal and vertical pumps and exhaust draft fans, this new piloted coupling combines the resiliency and shock cushioning action of the Falk general duty Steelflex coupling with a low angle universal joint action. This construction eliminates the necessity for an outboard bearing on extended shafts, or auxiliary bearings on intermediate shafts. The piloted coupling also allows greater flexibility in locating the driving and driven units.



Constructed entirely of steel, it consists of two hubs, one of which carries an integral flange to which the cover is secured; a special tempered steel member forming a complete cylindrical grid-member; and a flanged steel cover secured to the hub flange with capscrews. Available in single piloted, double piloted horizontal, and double piloted vertical. The double piloted drives are recommended where there considerable distance between the driving and driven equipment. The Falk Corp., Milwaukee, Wis.

Flexible Tube Fittings

New tube fittings, to be known as "Sealflex", will handle high pressure and will permit vibratory flexing up to 10 deg. included angle. They will reduce sound transmission in lines and permit rapid assembly without special tools for the lower capacity uses. The high pressure capacity and the flexibility of the tube-to-fitting connection is the result of the unique design which uses an insulating flexing medium of



rubber or synthetic rubber. The tubing is prevented from slipping by a metal ferrule which is "rolled in" on it. Thus the connection does not depend upon the shear strength of the rubber, and eliminates the possibility of pressure swedging down the end of the tube. Restriction in the line when the ferrule is "rolled in" to handle highest pressure causes a restriction of only 0.002 in. on the inside diameter. Available in a wide range of standard sizes for handling virtually any fluid. Harris Products Co., 5428 Commonwealth Ave., Detroit, Mich.

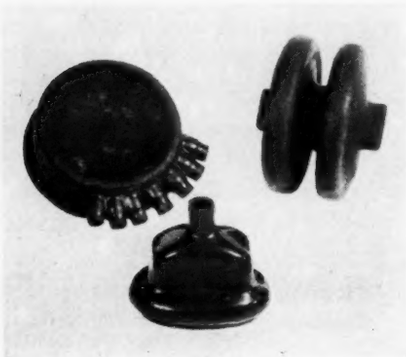
Rust Preventive Paint

Totrust rust preventive oil paint has a unique oil vehicle which penetrates the pinholes and rusted pits of the metal, expels any moisture present, seals in and around the rusted particles to form a hard, durable, but flexible film. A light gray color of Totrust is available which has an appearance and light reflection strikingly similar to aluminum and can be used as a sub-

stitute for aluminum paint at about one-third the cost. Other colors available are gloss black, red, machine gray, olive green, white and clear. The paint is made from titanium-base pigment. Drying time is from 8 to 12 hr. Totrust is also an excellent water proofing material for masonry or wood. Wilbur & Williams Co., Park Square Bldg., Boston, Mass.

Flexible Boots

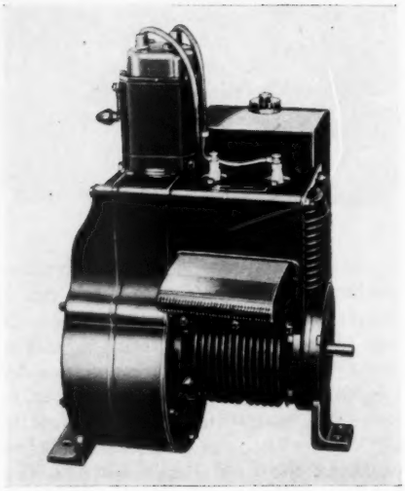
Flexible boots used as dust and moisture sealing covers, corrosion resistant protecting caps, and other rubber parts requiring a uniform thin wall up to one-eighth inch in thickness, are being produced from both rubber and neoprene by a process known as form-dipping.



Most rubber parts designed with a uniform thin wall lend themselves more readily to this type of production than any other process. Experimental production parts can be made on inexpensive forms in various given wall-thickness dimensions to determine which is the most suitable. Production forms are comparatively inexpensive, running in most instances but a fraction of the cost of a steel mold. Parts having maximum elasticity and flexibility, with tensiles up to 5,500 lb. per sq. in. are being produced by this process which has found unusual acceptance in the automotive industry. Automotive Rubber Co., 8601 Eyworth Blvd., Detroit, Mich.

Portable Engine

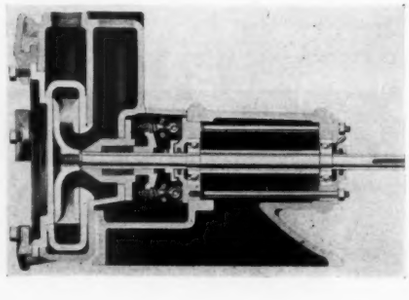
Three new "Mercury" 2-cycle, portable, air-cooled industrial engines have demonstrated, in tests, easy starting, unusual freedom from carbon and spark-plug trouble; cool running after many hours of continuous use, and have fewer and simpler working parts. The line consists of a 1-cylinder, 2½-hp. engine; a 2-cylinder, 5-hp. engine; and a 4-cylinder model which develops 10 hp. The 5-hp. model with clutch and



transmission weighs only 65 lb. Motors can be used in either vertical or horizontal position, and are available complete with clutch, various gear ratios, and with swivel transmission. Industrial type magnetos instead of the usual fly-wheel type are used. The engines may be throttle or governor-controlled, depending upon the application. Kiekhaefer Corp., Cedarburg, Wis.

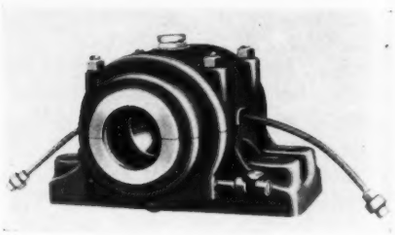
Horizontal Pumps

Designated types A and T, two new Amsco-Nagle horizontal industrial pumps are now available in all sizes from ¾ to 6 in., with capacities up to 1,400 g.p.m., and for operating heads up to 100 ft. These pumps are constructed of material best suited for the conditions of abrasion or corrosion under which the pump will operate. Metals specified where necessary and available are manganese steel, chromium-nickel alloys, white iron, nickel-iron, brass, bronze and other cuprous alloys. Features of the new pumps are: clamp-bolt construction of the water end; a bearing arrangement that assures proper alignment; sealed bearings; a readily accessible stuffing box; and self-priming. Four types of impellers are available: 2, 4 and 6-vane shrouded type and 4-vane open type. American Manganese Steel Div., American Brake Shoe & Foundry Co., Chicago Heights, Ill.



Precision Pillow Blocks

Designed for a wide range of industrial applications where a high grade babbitted bearing is required, new Sleevoil precision pillow blocks in plain and water-cooled types are available for shaft sizes ranging from 1 7/16 in. to 8 in. dia. Ample lubrication is provided by T-section brass oiling rings. Single rings are furnished in sizes up to 3 7/16 in. and two rings are used in larger sizes. A brass oil gage is provided



which can be placed on either side of the bearing. Bases have drain holes in each end. The bearings are fully self-aligning and can be furnished in either expansion or non-expansion types. Effective sealing against dirt is provided by special seals held onto the shaft by garter springs so that they revolve with the shaft and give the effect of piston ring seals. Dodge Manufacturing Corp., Mishawaka, Ind.

Synthetic Rubber

A new type of neoprene synthetic rubber that withstands the effects of sub-zero temperatures better than other types of neoprene, combines the desirable qualities of natural rubber at low temperatures and the oil-resistance qualities characteristic of all types of neoprene. Up to the present all known types of synthetic rubber have hardened when exposed to extremely low temperatures. In addition, the vulcanizates of Neoprene Type FR are more resilient, take lower compression set, and have greater resistance to swelling by water than those of other neoprenes. Like rubber, the Type FR Neoprene requires sulfur to obtain an adequate state of cure in normal vulcanizing operations. Like other types of neoprene, it has excellent flame and sunlight resistance. It is supplied in the form of preplasticized, milled sheets and has a specific gravity of 1.15—lower than any other neoprene. Principal early applications will be in the aircraft and automotive field for devices which must operate satisfactorily at either high or low atmospheric temperatures. E. I. du Pont de Nemours & Co., Inc., Wilmington, Del.

Manufacturers' Publications

AVAILABLE PROMPTLY UPON REQUEST ADDRESSED DIRECTLY TO THE MANUFACTURER

Materials

ALUMINUM — Aluminum Company of America, Pittsburgh, Pa. Aluminum Service Bulletin, 8½x11 in. In order to serve defense industries with up-to-date technical information, this new series of service bulletins is now being issued. Aluminum Service Bulletin No. 1 lists 100 references on aluminum.

AMPCOLOY—Ampco Metal, Inc., 1745 S. 38th St., Milwaukee, Wis. Catalog, 15 pages, 8½x11 in. In addition to general description of the Ampcoloy high quality bronze alloys, a comprehensive table of physical properties and a range of chemical compositions is included. The booklet is well illustrated.

AMPCO METAL—Ampco Metal, Inc., 1745 S. 38th St., Milwaukee, Wis. Engineering Data Sheet File 41, 56 pages, 8½x11 in. The most important of the Ampco data sheets that have been published since 1934 have been selected and published in this booklet, covering a wide range of subjects.

DUREX IRON — Moraine Products Div., General Motors Corp., Dayton, Ohio. Bulletin, 6 pages, 8½x11 in. Powder metallurgy as a practical answer to the production of bearings, gears and cams, and miscellaneous metal parts, is discussed in this bulletin.

FELTS — Booth Felt Co., 434-19th St., Brooklyn, N. Y. "The Technique of Felt Making," 24 pages, 8½x11 in. Complete engineering and design data for felt is included in this booklet, as well as the comprehensive story of its raw materials and manufacture. Two pages of bearing diagrams with felt seals, and the A.S.T.M. and S.A.E. specifications covering felt are included. The latter specifications include a list of recommended uses.

NICKEL ALLOYS — International Nickel Co. Inc., 67 Wall St., New York, N. Y. "Individualized Inco Nickel Alloys," 12 pages, 8½x11 in. Gives characteristics, mechanical properties and application information on the Inco nickel alloys, including monel, nickels, "Z" nickel, and Inconel. Tables of physical constants and available forms are included.

SPRING WIRE—Chicago Coil Spring Co., Albany & Carroll Aves., Chicago, Ill., 3¾x8½ in. Contains samples and gives description and comparison of five principal types of spring wire, including music wire, oil tempered wire, high carbon wire, phosphor bronze spring wire, and brass spring wire.

WIRE ROPE — MacWhyte Co., Kenosha, Wis. Catalog G-14, 166 pages, 4¾x6½ in. This catalog is veritable encyclopedia of wire rope information. In addition to the well-presented pages of catalog listings, many pages in the back of the book give all the engineering data necessary for specifying the proper wire rope. Sheave design and maintenance, inspection, and manufacture are covered. Tables of general data are also included.

Mechanical Parts

CONE-DRIVE GEARING—Michigan Tool Co., 7171 E. McNichols Rd., Detroit, Mich. Bulletin CW 41B, 12 pages, 8½x11 in. This booklet contains considerable information not previously published, including illustrations of numerous installations, details on how cone drives are produced, specifications of speed reducers, and a discussion of gearing in general.

HYDRAULIC DRIVES — Twin Disc Clutch Co., Racine, Wis. Bulletin 132, 24 pages, 8½x11 in. This well-prepared and comprehensive description of hydraulic clutches and hydraulic torque converters contains much general engineering information on design and performance.

SPRAY NOZZLES—Spraying Systems Co., 4011-21 W. Lake St., Chicago, Ill. 1941 catalog describes and illustrates complete line of spray nozzles for all industrial uses, and contains many performance tables and data sheets.

VALVES—Hills-McCanna Co., 3025 N. Western Ave., Chicago, Ill. 1941 catalog, 8½x11 in. Comprehensive catalog illustrates and explains the outstanding features of diaphragm valves, including factual data and specifications and service recommendations.

VIBRATION INSULATORS—B. F. Goodrich Co., Akron, Ohio. Catalog Section 7900, 12 pages, 8½x11 in. This excellently illustrated booklet gives a general discussion of vibration insulation in machines, describes Vibro-Insulators, gives methods of selecting, and discusses uniformity of load distribution, rocking and horizontal motion and the care of the devices.

Electrical Parts

MOTORS—Allis-Chalmers Mfg. Co., Milwaukee, Wis. Bulletin B-6052-B, 8 pages, 8½x11 in. The company's Lo-Maintenance motors in ratings from ¾ to 75 hp., open, inclosed and splash-proof types, a.c. and d.c. are concisely described and illustrated in this bulletin. Types of motors and their applications are summarized in tabular form.

MOTORS—General Electric Co., Schenectady, N. Y. Bulletin GEA-3579, two pages, 8x10½ in. This leaflet gives a brief description of fractional-horsepower motors for machine tools and other industrial applications.

PANEL INSTRUMENTS—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Catalog Section 43-330, 12 pages, 8½x11 in. Miniature a.c. and d.c. voltmeters and ammeters in the 2-in. specifications for general use are described. A complete tabulation lists ratings, scale divisions, style numbers, and list prices.

PRECISION SWITCHES — Micro Switch Corp,. Freeport, Ill. Catalog page 5.021, 8½x11 in. Announces a reduction in the list price and an extension of the discount on Type H Micro Aircraft Housings and Switches.

RESISTORS — Keystone Carbon Co. Inc., 1935 State St., St. Marys, Pa. Bulletin, 4 pages, 8½x11 in. Keystone negative temperature coefficient resistance material is described in this new folder. The last page contains a curve which indicates the variation of resistance with temperature.

RESISTORS—Sprague Specialties Co., Resistor Div., North Adams, Mass. Koolohm Resistor Catalog, 8½x11 in. Illustrates and describes the complete line of Koolohm resistors, including many new types. Detailed information on this resistor construction is included.

Miscellaneous

DEFENSE—Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill. Book 1917, 12 pages, 8½x11 in. Entitled "Just Another Job," this interesting illustrated booklet describes the work Link-Belt is doing directly or indirectly for National Defense.

SILVER BRAZING—Handy & Harman, 82 Fulton St., New York, N. Y. Bulletin 12, 16 pages, 18½x11 in. This comprehensive booklet gives a general description of Sil-Fos and Easy-Flo silver brazing alloy. Of particular interest to product engineers, several pages of applications are given and many sketches show the proper method for designing silver-brazed joints. Formulas for calculating shear depth and other engineering factors are included.

Engineering Department

TACHOMETERS—Boulin Instrument Corp., 65 Madison Ave., New York, N. Y. Bulletin, 4 pages, 8½x11 in. Sets forth features and specifications for Boulin centrifugal tachometers of the single range hand, the stationary, and the triple range centrifugal types.

TESTING MACHINES—Baldwin-Southwark Div., Baldwin Locomotive Works, Philadelphia, Pa. Bulletin 161, 40 pages, 8½x11 in. Illustrated with more than 50 photographs and charts, this bulletin pictures the company's testing machines in capacities from 20,000 to 4,000,000 lb. General specifications and a detailed description of operation are included for all types.

Books and Bulletins

Photoelasticity

MAX MARK FROCHT—*Volume I, 411 pages, 5½x9 in. Black clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $6.*

Based almost entirely upon researches conducted by the author and staff of the Photoelastic Laboratory of the Carnegie Institute of Technology, as well as the author's teaching experience, this volume contains the essential material for a thorough understanding, by any engineer, of the theoretical principles and experimental procedures necessary for the complete exploration of all two-dimensional stress systems by the method of photoelasticity. Engineers who have retained a working knowledge of differential and integral calculus and strength of materials will have no difficulty in reading this text.

The scope of the book is limited to fundamental principles and methods, and in the main confines itself to two-dimensional problems. The methods for plane stress analysis dealt with in this volume are based entirely on the stress pattern and the isoclinics.

Engineers who are interested in the methods by which reliable non-destructive quantitative stress analyses of structural elements can be made will find this volume to be of great value.

Industrial Plastics

HERBERT R. SIMONDS—*Second Edition, 397 pages, 6½ x 9¼ in. Illustrated and indexed. Green clothboard covers. Published by Pitman Publishing Corp., 2 West 45th St., New York, N. Y. Price $4.50.*

Since progress in plastics is rapid, constant revisions are always necessary to keep any book on this subject abreast of the field. Recognizing this, Mr. Simonds has carefully gone over his original text, published in 1939, (reviewed in the November 1939 issue of PRODUCT ENGINEERING, page 507). All tables and statistics have been brought up to date, the chapter on chemistry re-written and formulas revised. Also included are data on new plastics materials and a chapter on plastics in ordnance.

Purposely, the author has principally concerned himself with the use of plastics as engineering materials. Thus the text is made up of comprehensive data on how to design with the various plastics materials, while of secondary consideration to the design engineer, the chapters on molding practices and plant equipment are valuable background material.

Engineering Encyclopedia

Two volumes. 1,431 pages, 6 x 9 in. 206 illustrations. Published by The Industrial Press, 148 Lafayette St., New York, N. Y. Price $8.

Packed into these two volumes is an amazing amount of mechanical engineering information, written in true encyclopedia style. Essential facts about 4,500 standard and special engineering subjects are concisely treated in a few paragraphs, or in a few pages, depending upon the importance of the subject. All subjects are arranged alphabetically, with cross references wherever necessary.

Various important mechanical laws, rules, and principles; physical properties and compositions of a large variety of materials used in engineering practice; the characteristic features and functions of different types of machine tools and other classes of manufacturing equipment, and countless other subjects are included. These condensed treatises are accompanied by definitions of numerous trade and technical terms used in engineering and manufacturing practice. Historical discussions add to the interest of the subjects.

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Chip Formation, Friction, and Finish

HANS ERNST AND M. EUGENE MERCHANT—*32 illustrations, 48 pages, 6x9 in. Published by Cincinnati Milling Machine Co., Cincinnati, Ohio. Available free.*

To anyone interested in the physics of metal cutting, this booklet will be of great interest. Mr. Ernst and Mr. Merchant in their extensive research have arrived at some quantitative relationships between chip formations, friction, and finish. Their research covered geometrical analysis of chip formation, friction investigations, photomicrographic examinations, and studies of the action of cutting fluids.

Their conclusion is that great advances in the field of metal-cutting may yet be made by developing means for further reducing the coefficient of friction between chip and tool. Additional research must be carried out to develop cutting fluids which produce low shear strength material at the chip-tool interface, even at very high speeds.

Quenched and Tempered Nickel Alloy Steels

Data Sheet Section II, No. 4, 104 pages, 6½x8¾ in. Published by International Nickel Co., Inc., 67 Wall St., New York, N. Y. Available free.

This revised section to International Nickel Company's "Nickel Alloy Steels" data book gives complete information covering the effect of quenching and tempering nickel steels. After a general discussion of the properties of the various S.A.E. specifications, the booklet presents many pages of mechanical property charts, graphs showing effect of size of section on the tensile properties of quenched and tempered nickel alloy steels, and charts showing the torsional properties of quenched and tempered nickel alloy steels in different sizes.

Textolite

64 pages, 8x10½ in. Published by General Electric Co., Pittsfield, Mass. Available free.

To those design engineers who want to pick up general information on phenol-formaldehyde and similar plastics, both molded and laminated, this introductory encyclopedia on plastics manufacturing is recommended. Each step in the manufacture is covered, from raw materials to the finished product.

Of specific interest to the design engineer are the chapters covering designing and engineering, mold making, and industrial designing. Properties of Textolite molded and Textolite laminated are presented in chart form in the two final chapters.

Design and Rating of Cone-Drive Worm Gearing

Catalog CW41A, 44 pages, 9x11 in. Published by Cone-Drive Div., Michigan Tool Co., 7171 E. McNichols Rd., Detroit, Mich. Available free.

This design and rating manual for design engineers includes complete specifications of standard blanks for worms and wheels, tables of all sizes of ratios for which tooling is available, and charts and formulas for computing stresses, loads, horsepower ratings, and gear proportions and design.

Crank Mechanism Factors

LEVI B. SMITH—*20 pages, tables, 6x9 in. Available from author, P.O. Box 317, Hampton, Va. Price 40 cents.*

Displacement, velocity and acceleration factors for reciprocating motion, based on exact formulas, are tabulated to fill a need for data over a wider range than the author has found available heretofore. Factors correspond to L/R ratios from 2 to 6, where L is connecting-rod length and R is crank length. All factors are given to four places and the increments are chosen to make interpolation linear for any given crank angle. Examples are worked out to show how the tables are used.

HELICAL SPRING SELECTION CHART

HANS A. ILLING

Definitions Applying To Round Steel Wire

D = mean diameter of coil in inches
d = wire diameter in inches

$$P_1 = \frac{100{,}000}{S} P$$

(P = safe load for a given stress S)

$$y = \frac{L P}{f}$$

(L = solid length of acting coils and f = total deflection

Basic Formulas

(1) $$f = \frac{8 N D^3 P}{G d^4}$$ where N = number of acting coils, $G = 11{,}500{,}000$

(2) $$P = \frac{\pi d^3 S}{8 D K}$$

(3) $$K = \frac{4c - 1}{4c - 4} + \frac{0.615}{c}$$ where $c = \frac{D}{d}$

K = Wahl's correction factor

Derivation of Equations for P_1-Curves

Solve Equation (2) for D, let $S = 100{,}000$ and substitute P_1 for P. The resulting equation is:

$$D = 12{,}500\,\pi \cdot \frac{1}{P_1} \cdot \frac{1}{K} \cdot d^3$$

which in the logarithmic form is:

$$\log D = 3 \log d + \log \frac{(12{,}500\,\pi)}{P_1} - \log K$$

Except for the correction factor K this is the equation for a straight line on logarithmic coordinates. It will be noted that the P_1 lines on the chart are not quite straight.

Derivation of Equations for y-Curves

Solve Equation (1) for D after substituting L/d for N and y for LP/f

$$D = 3\sqrt{\frac{11{,}500{,}000}{8y}} \cdot d^{5/3}$$

which in the logarithmic form is:

$$\log D = \frac{5}{3} \log d + \frac{1}{3} \log \frac{11{,}500{,}000}{8y}$$

This is the equation for a straight line on logarithmic coordinates. Variations of y give the set of y-lines on the chart.

Applications of the Helical Spring Chart

(1) Round Steel Wire:

Only the equations for P_1 and y need be used with the chart. Example:

P = 160 lb.
S = 80,000 lb. per sq. in.
f = 2 in. (under load of 160 lb.)

The compressed length of the spring must be 5 in., from which it is estimated $L = 3.75$ in., allowing 1.25 in. for the two end coils and for clearance between coils.

Then $$P_1 = \frac{100{,}000}{80{,}000} \cdot 160 = 200 \text{ lb.}$$

$$y = 3.75 \cdot \frac{160}{2} = 300$$

The intersection of these P_1 and y lines are at the point where $D = 1$ in. and $d = 0.185$ in. (approx. $\frac{3}{16}$ in.).

(2) Round Wire of Other Material:

Proceed as above except that the y equation becomes:

$$y = \frac{11{,}500{,}000}{G} \cdot 3.75 \cdot \frac{160}{2} = \frac{11{,}500{,}000}{G} \cdot 300$$

If $G = 10{,}500{,}000$ $\quad y = 328.5$, $P_1 = 200$

$D = .96$ and $d = .182$

(3) Square Steel Wire:

$$P_1 = \frac{94{,}250}{S} \qquad P = \frac{94{,}250}{80{,}000} \cdot 160 = 188.5 \text{ lb.}$$

$$y = 0.695\ L\frac{P}{f} = 0.695 \cdot 3.75 \cdot \frac{160}{2} = 208.5$$

(assuming d becomes the thickness of square section)

$D = 1.1$ in. $\quad d = 0.1825$

(4) Square Wire of Other Material:

$$P_1 = \frac{94{,}250}{S} P \text{ (same as above)}$$

$$y = \frac{8{,}000{,}000}{G} L\frac{P}{f} = \frac{8{,}000{,}000}{G} \cdot 300$$

Using the same assumed values as before

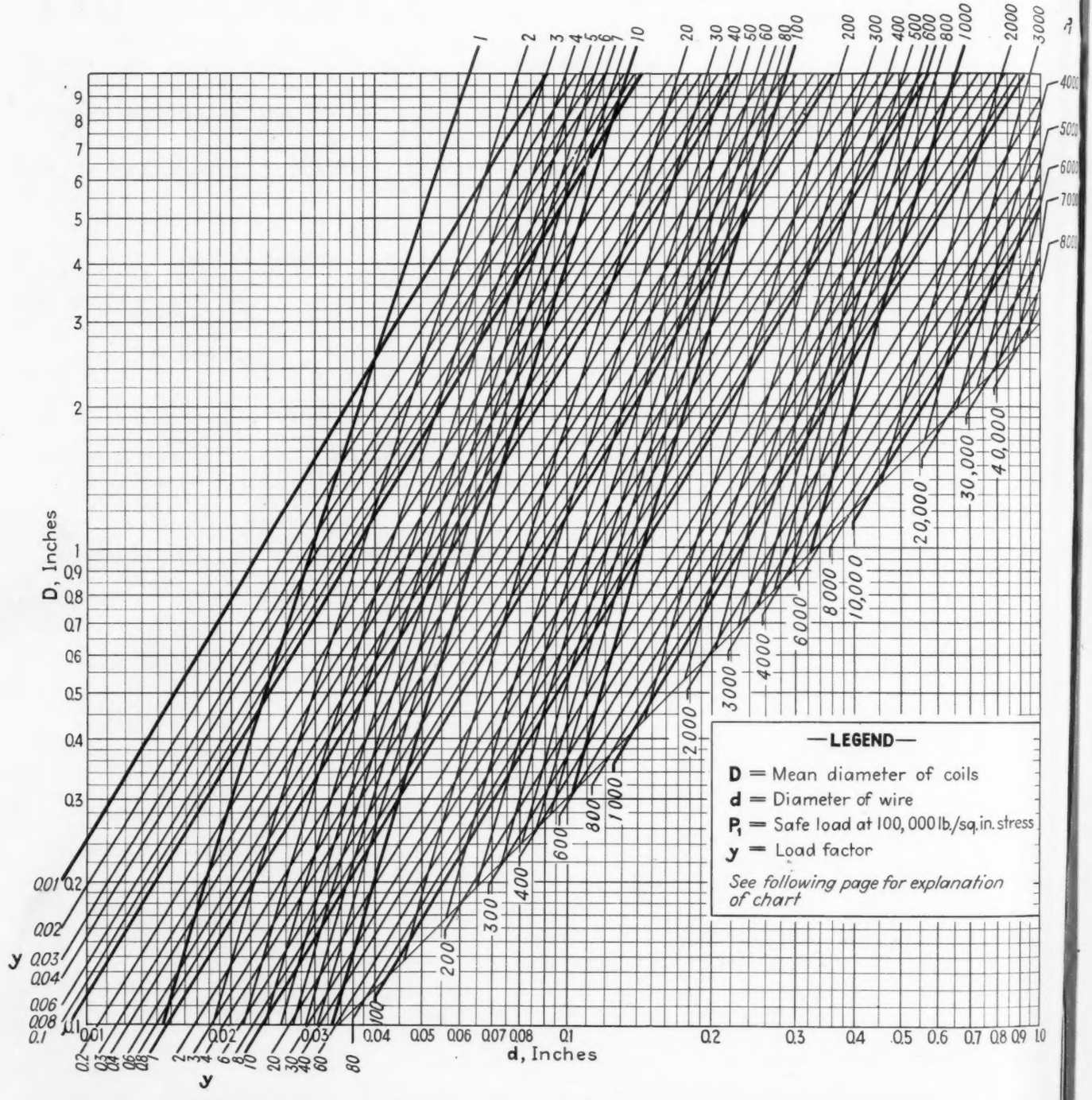
$$P_1 = \frac{94{,}250}{80{,}000} \cdot 160 = 188.5 \text{ lb.}$$

$$y = \frac{8{,}000{,}000}{10{,}500{,}000} \cdot 3.75 \cdot \frac{160}{2} = 228.5$$

$D = 1.125$ and $d = 0.185$

HELICAL SPRING SELECTION CHART

HANS A. ILLING



PRODUCT ENGINEERING ✦ REFERENCE BOOK SHEET